Copyright, 1935, by the AMERICAN SOCIETY OF NAVAL ENGINEERS.

JOURNAL

OF THE

AMERICAN SOCIETY OF NAVAL ENGINEERS

VOL. 47. AUGUST 1935. No. 3

The Society as a body is not responsible for statements made by individual members

COUNCIL OF THE SOCIETY

(Under whose supervision this number is published).

Admiral WILLIAM H. STANDLEY, U.S.N. — Commander H. N. PERHAM, U.S.C.G.
Captain J. S. EVANS, U.S.N. — Lieut. Comdr. E. W. MILLS, U.S.N.
Captain B. H. BRUCE, U.S.N. — Mr. JOHN F. NICHOLS
Lieut. Comdr. W. C. WADE, (C.C.) U.S.N. — Mr. W. M. CORSE
Commander C. S. GILLETTE, U.S.N.

FOREWORD.

BY THE SECRETARY.

It is the established policy of the Council of this Society to confine the selection of the majority of articles for its JOURNAL to those of primary interest to the operating engineer, whenever suitable material is available.

For this issue, however, it has seemed best to depart from this policy and to publish in detail the results of an extensive study and research recently completed at the U. S. Naval Engineering Experiment Station, which are considered to be of primary interest to the marine power plant designer and to the designer of power plants on shore as well. Nevertheless, the operating engineer may find much of interest in the designer's uncertainties about the calculations of pipe line stresses for an actual installation which may indicate the answer to some of his maintenance difficulties.

It is not always the workmanship or the material which is at fault in producing the leaky joint, but frequently the design of the piping layout which produces operating stresses beyond the ability of the

material used to withstand. Not only is the calculation of such stresses a difficult and complex matter, but the behavior of materials at high temperatures is still a somewhat uncertain problem, particularly as to the behavior of steel under cyclic stress at elevated temperatures.

The modern trend towards higher steam pressures and temperatures demands that the operating engineer have at his disposal more concise information in regard to the tools which he must employ than ever before. The use of corrugated and creased piping is of great importance, especially in the confined spaces available for the power plant installation on board the modern fighting vessel. Realizing this, and noting the lack of concise information in regard to the practical use of this type of piping, the Bureau of Engineering, Navy Department, authorized the necessary tests on full-size specimens, which were generously contributed by the manufacturers of this class of material. The results given in the principal paper published in this issue will, it is believed, be of great interest to power plant designers universally.

For those unfamiliar with the general subject of corrugated and creased piping it is suggested that an article from "Engineering" of 16 December 1932 on "High Pressure and High Temperature Steam Pipe Work," herein reprinted under the general heading of "Notes," might be read with profit before study of the main paper is undertaken.

The Bureau of Engineering now requires, by its specifications, complete stress calculations of pipe line layouts and restricts very definitely the maximum calculated stress allowable. This should result in a material decrease in pipe line maintenance. However, the problem for calculation is inherently complex and the primary reliance must be a conservative attitude in the selection of the factor of safety to be used. Further, as indicated by installations now in service, a completed pipe layout probably tends to adjust itself to relieve the points of maximum stress, and so to produce more uniform stress conditions within the capacity of material to withstand. The general problem can best be met by carefully calculated layouts of piping, the use of conservative factors of safety, and the employment of better materials of higher endurance limit and creep-resistant characteristics at the temperatures now used in service.

THE STRENGTH AND FLEXIBILITY OF CORRUGATED AND CREASED-BEND PIPING.

By LIEUT. ROBERT L. DENNISON*, U. S. NAVY, MEMBER.

I. INTRODUCTION.

1. *The General Problem of Design Involving High Temperatures and Pressures.*—The present day trend in the direction of higher steam temperatures and pressures as a means of increasing thermal efficiencies of steam cycles has necessitated the solution of many complex problems of engineering design. It may be said that the greatest barriers to the immediate utilization of the steam pressures and temperatures which theory indicates as being so desirable, are the lack of suitable metals and the lack of full knowledge of the behavior of available materials under conditions of temperature, stress, erosion, and corrosion. However, in many cases when the limitations of material are sufficiently known, the design is uncertain because of failure to foresee accurately under what conditions the material is to function in service. If the forces acting and the manner in which the act were known, if the stresses and strains which these forces produce in the material could be determined, and if the physical limitations of the material under the environmental conditions were fully appreciated, the optimum design could readily be accomplished. Ignorance of one or several of these design factors leads to the use of a so-called factor of safety. The machine element is designed so that the highest predicted stress multiplied by the factor of safety shall not exceed the yield strength or the ultimate strength, whichever is used as the basis for design. In some cases the fatigue limit is substituted for the yield strength or ultimate strength as the basis for the factor of safety, and at least one system of determining the allowable working stress is based on a combination of yield strength and fatigue

* Formerly attached to the U. S. Naval Engineering Experiment Station, Annapolis, Maryland.

strength*. The demands of low cost, minimum weight, and least space provide an ever increasing temptation to reduce the factor of safety. In order to do this intelligently, it is vitally necessary to possess full knowledge of all factors entering into the design and of those conditions in service which affect the strength of the material.

2. *The Design of Piping Systems.*—A typical problem exists in the design of steam piping. Increasing temperatures means that additional expansion must be absorbed. With high steam temperatures and long runs of piping, the expansion to be absorbed may be considerable. Four means for caring for pipe line expansion are generally accepted; swivel joints, expansion joints, slip joints, and the configuration of the piping system itself. Swivel joints are unsuitable except for low pressures and temperatures and are only used with small pipe sizes. The expansion joint is capable of absorbing little movement and is generally used under conditions where some materials such as rubber or copper might be successfully employed. Slip joints involve the use of packing and stuffing boxes. Because of leakage and uncertainty of action, they are seldom used at temperatures exceeding 400 degrees F. For high steam temperatures and pressures it is necessary to design a piping system so that the expansion may be absorbed by the deformation of the pipe line itself. This is usually accomplished by the inclusion of expansion bends or by changes in direction in the run of piping. It will be appreciated that where the expansion is taken up by the piping system, reactive forces will be exerted on the anchorages equal to those required to cause the necessary deflections in the piping. High reactive forces on the turbine flanges or boiler drums may cause serious distortion of these units. It is therefore desirable to have a high degree of flexibility in the piping system and hence small forces resulting from expansion.

3. *The Need for Flexible Piping.*—Because of the high reactive forces associated with the usual expansion elements, it became necessary to employ several such elements in runs of piping of moderate length. Where the steam piping changed direction large arcs were necessary because the usual piping requires the maintenance of a relatively large ratio of radius of curvature of the bend to the

* "Strength of Materials," Vol. 11, S. Timoshenko.

pipe diameter so as properly to form the bend. In order to provide increased flexibility and to permit forming of bends on a small proportion of the arc radius to pipe diameter, the so-called corrugated piping and creased-bend piping were developed. Typical examples of these pipe forms are shown on Plates IX and XI. These forms are in commercial use here and abroad.

4. *Objects of This Investigation.*—This investigation had the following objects: First, to determine the relationship between the load and deflection for various shapes of corrugated and creased-bend pipes in order to establish their flexibility relative to smooth-walled pipe; second, to determine the endurance properties of the corrugated and creased-bend pipes under cyclic stressing; third, to develop methods for measuring the strains induced in these pipes under load and interpreting the significance of these strains; and finally, to derive values for safe working stresses for these pipes for the purpose of piping system design.

II. PREVIOUS WORK.

5. *Early Experimental Work on Expansion Bends.*—In order to trace the development of the analysis of the strength and flexibility of pipe bends, it is desirable to review briefly the work of other investigators, not only on the creased and corrugated bends, but on smooth pipe bends as well. Perhaps the earliest important work was done by Dr. A. Bantlin of Stuttgart, who experimented with the lyre-shaped expansion bend,* sometimes called the double offset expansion U-bend. These experiments on smooth pipe showed that in the case of expansion bends formed of mild steel pipe the displacement of the ends when subject to forces tending to alter the curvature might be as much as five times greater than could be predicted by the ordinary theory of bending of curved bars. Dr. Bantlin concluded that the discrepancy between theory and measurement could be ascribed to a virtual reduction in the flexural modulus of rigidity, EI. This virtual reduction was believed to be caused by a large number of minute waves or folds on the inner side of the pipe bend which were formed in the bending of the pipe. Later, of course, the creased-bend pipes and corrugated pipes were designed

* "Formänderung und Beanspruchung federnder Ausgleichröhren," by A. Bantlin Zeitschrift des Vereines deutscher Ingenieure, January 8, 1910, p. 43.

in which folds were deliberately produced in order to augment flexibility.

6. *Development of the Theory of Flexibility of Pipe Bends.*—Dr. Th. V. Karman of Göttingen later analyzed the results of Dr. Bantlin's experiments and advanced the hypothesis that the discrepancies noted between measurement and theory were due to the deformation of the circular pipe section when the curvature of the bend was changed.*

It will be appreciated that when a pipe bend is subjected to couples acting on the ends tending to increase the curvature, the tensile forces in the pipe wall on the outside of the arc and the compressive forces in the pipe wall on the inside of the arc, lying in the plane of the bend, will each have components tending to force the walls in towards the neutral axis of the pipe. These components will tend to form the cross-section into a quasi-elliptical shape. This deformation will lessen the value of the moment of inertia of the cross-section of the pipe and will hence operate to increase the flexibility. The greatest cause of increased flexibility is, however, the relieving of stresses in the fibers farthest from the neutral axis of the pipe as a result of the flattening, with a consequent increase of the angular deflection of the bend before equilibrium of the internal resisting moment and external bending moment will occur. The effect on flexibility is equivalent to an increase in value of the bending moment or a decrease in the value of the moment of inertia of the cross-section of the pipe.

Hence, in finding the deflection of a curved pipe the value of flexural rigidity, EI, must become EI/K, where K is a dimensionless factor greater than unity.† K is expressed as a function of λ, where

$$\lambda = \frac{tR}{r^2} \text{ and}$$

* "Ueber die Formänderung dünnwandiger Rohre, insbesondere federnder Ausgleichröhre," by Th. V. Karman, Zeitschrift des Vereines deutscher Ingenieure, 11 November 1911, p. 1889.

† This factor, K, is defined by some authors as a factor less than unity and by others as the reciprocal of this value. As originally derived by Dr. Karman the factor had a value less than unity. The writer prefers to refer to this factor as having a value greater than unity. This convention will be followed in this paper in all cases where K appears.

t = thickness of pipe wall
R = mean radius of pipe bend
r = mean radius of cross-section of pipe
then

$$K = \frac{10 + 12\lambda^2}{1 + 12\lambda^2}$$

This expression for K is closely approximate for values of λ greater than 0.3. Dr. Karman applied this theory to the results of Dr. Bantlin's experiments and obtained close agreement.

7. *Researches at Massachusetts Institute of Technology.*—This important work by Dr. Karman did not attract much attention until Professor W. Hovgaard of the Massachusetts Institute of Technology undertook extensive investigations of the flexibility and strength of pipe bends.* Using an entirely different method of analysis, Professor Hovgaard proved the correctness of Dr. Karman's derivation of the value of K and checked the theory with observations on several pipe bends. The relieving of stresses in the fibers most distant from the neutral axis of the pipe in the plane of bending causes the longitudinal stresses not to follow a linear law but provides maximum values of stress nearer to this axis. Formulas for determining the value of the longitudinal stress were developed by both Karman and Hovgaard and were confirmed experimentally by the latter. It was also shown both experimentally and theoretically by Hovgaard that the transverse, or hoop, stresses which occurred at the ends of the major and minor axes of the deformed cross-section might, under certain circumstances be much greater than the longitudinal stresses.

8. *The Strength of Pipe Bends.*—The tests at the Massachusetts Institute of Technology were made with the pipes under static loading conditions only. A pipe bend was considered to have failed when its elastic range was exceeded. Hence, the object of the test was to discover the state of stress at which the pipe bend

* "The Elastic Deformation of Pipe Bends," by W. Hovgaard, Journal of Math. & Phys., Mass. Inst. of Tech., Vol. VI, No. 2, 1926.

"Deformation of Plain Pipes" and "Further Research on Pipe Bends," by W. Hovgaard, Journal of Math. & Phys., Mass. Inst. of Tech., Vol. VII, No. 3, October, 1928, and No. 4, December, 1928.

"Tests of High Pressure Pipe Bends," by W. Hovgaard, Journal of Math. & Phys., Mass. Inst. of Tech., Vol. VIII, No. 4, 1929.

would suffer a permanent set. This point was then taken as determining the limit of strength of the bend. The point of limiting strength, therefore, was taken at the limit of plasticity. It is important to appreciate this basis of estimating strength. Professor Hovgaard investigated the maximum strain energy theory according to Dr. M. T. Huber and Dr. H. Hencky, as defining the criterion of failure.* According to this theory the limit of beginning of plastic flow depends on the elastic work of deformation. When this work reaches a certain constant value at a given point, flow begins. The work of pure elastic deformation is given by the formula—

$$W = \frac{m + 1}{3\,m\,E}\,(p_1^2 + p_2^2 + p_3^2 - p_1p_2 - p_2p_3 - p_1p_3)$$

where p_1, p_2, p_3 are the principal stresses and m is the reciprocal of Poisson's ratio.

If W is constant its value at the yield point in one-axial tension may be expressed

$$W = \frac{m + 1}{3\,m\,E}\,p_0^2$$

where p_0 = the value of tensile stress at the yield point. Hence, in bi-axial stress

$$p_1^2 - p_1p_2 + p_2^2 = p_0^2$$

Hovgaard now assumes

$$p_e = (p_1^2 - p_1p_2 + p_2^2)^{\frac{1}{2}}$$

in which p_e is called equivalent stress.

Hovgaard found by analysis that, long before an appreciable permanent set occurred in the bends, this equivalent stress exceeded the yield point in certain sections of the pipe. These areas of high stress and plastic flow were considered to be localized and hence of not appreciable effect on the elasticity of the pipe as a whole. The longitudinal stresses in the outer surface of the pipe bend

* "Die spezifische Formänderungsarbeit als Mass der Anstrengung eines Materials," by M. T. Huber, Lemberg, 1904.

"Zur Theorie plastischer Deformation," Zeitschrift f. angew. Math. and Mech. Bd. 4, Heft 4, 1924, by H. Hencky.

appeared to reach the elastic limit at about the same time the permanent set began to appear. It was then assumed that the longitudinal stress determined the point of breakdown of pipe bends, and a formula is given for the determination of the maximum value of this stress. As long as this stress did not exceed about 20,000 pounds per square inch in the bends tested there was no permanent set. It was assumed that the yield point was about 25,000 pounds per square inch. Hovgaard made no distinction between "yield point" and "proportional elastic limit." The equation for the maximum longitudinal stress may be expressed

$$p_{max} = \beta \frac{Mc}{I}$$

$$\text{where } \beta = 2/3\ K \left(\frac{6\lambda^2 + 5}{18}\right)^{\frac{1}{4}}$$

A curve is presented by Hovgaard for determining values of β for various values of λ. In closing the investigation, he recommended that, because of overstrains which will exist in certain parts of the pipe which might in time produce a permanent set, the designed working stress be taken as 16,000 pounds per square inch.

9. *Work of Other Investigators.*—Mr. A. M. Wahl carried out some investigations tending to confirm the work of Dr. von Karman and Professor Hovgaard.* Mr. Wahl considered the longitudinal and transverse stresses induced by bending of the pipe to be principal stresses, and proposed that the maximum shear theory be used as a means for determining the critical stress.† By this theory the strength depends on the maximum shear stress which is one-half the difference between the greatest and least principal stresses. The tensile stress equivalent to this shear stress will be equal to the difference between these principal stresses. Mr. Wahl developed a means for expressing this equivalent stress as a function of the longitudinal stress. He provided curves using Karman's factor as an argument to assist in the computation. Wahl also concluded that the internal pressure has no appreciable effect

* "Stresses and Reactions in Expansion Pipe Bends," by A. M. Wahl, Trans. Amer. Soc. Mech. Eng. FSP-50-49, 1927.

† J. J. Guest, Phil. Mag., Vol. 50, 1900, p. 69.

on the tendency of the cross-section of the bends to become elliptical under a bending moment. Mr. J. R. Finniecome, an English engineer, made a critical survey of the work on stress and deflections of pipe bends and concludes with Mr. Wahl that the maximum shear theory should apply.*

10. *Fatigue Failures of Piping in Service.*—In 1914 Mr. C. E. Stromeyer read a paper before the Institute of Naval Architects commenting on the cause of failure in the cases of about one hundred steam pipe ruptures in service.† The pipes which failed were of copper and were of various shapes. The failures occurred after the pipes had been in service from 24 hours to 10 years. Mr. Stromeyer, in this early work, attempts to estimate the fatigue stresses to which the individual pipes were subjected. At this time little was known about endurance properties of metals and the methods of calculating stresses in pipe bends. This work, however, indicates an important point which appears to have been overlooked in later work on determining limiting design stresses for pipe bends, namely, that service failures are associated with cyclic, rather than with static, stress application.

11. *The Flexibility of Corrugated and Creased Pipe Bends.*—Messrs. E. T. Cope and E. A. Wert have conducted numerous tests on various shapes and sizes of corrugated and creased pipe bends. These tests were for the purpose of determining the flexibility characteristics and were conducted with various types of loading.‡ They concluded that the flexibility of the corrugated pipe bend is due mainly to a change in tangential length, increasing on the outside of the bend and decreasing on the inside of the bend, while little flattening of the cross-section occurs. The creased bend appears to show somewhat less flattening of the cross-section than a smooth bend and owes its flexibility both to flattening and to the change in tangential length between creases. They determined the relationship between the measured flexibility of the corrugated and creased pipe bends and the calculated flexibility of smooth

* "The Flexibility of Plain Pipe," by J. R. Finniecome, The Engineer (England), August 17, 1928, p. 162; August 24, 1928, p. 199; August 31, 1928, p. 218; September 7, 1928, p. 246.

† "The Elasticity and Endurance of Pipe Lines," by C. E. Stromeyer, Engineering (England), June 19, 1914, p. 856.

‡ "Load Deflection Relations for Large, Plain, Corrugated and Creased Pipe Bends," by E. T. Cope and E. A. Wert, Trans. A. S. M. E., 1931.

pipe bends of similar shape, dimension, and under similar loadings. Messrs. E. A. Wert, S. Smith and E. T. Cope have developed a series of graphs for the purpose of determining the correct proportions for ten of the most commonly used shapes of pipe under various operating conditions.*

III. THE DESIGN OF PIPING UNDER CONSIDERATIONS OF VARIABLE STRESS.

12. *Operating Conditions of Steam Pipes in Service.*—There are many ways by which a steam pipe may be stressed in service. In superheat systems the variation in steam temperature causes a corresponding variation in pipe line expansion with consequent variable stress. Changes in steam pressure will likewise create stress cycles. The piping system is subjected to a stress cycle each time it is placed in and out of operation. In addition to these, mechanical vibration of a piping system will induce cyclic stresses which, even though of small magnitude, superimposed on an existing high state of steady stress may lead to failure. On shipboard the weaving of the ship's structure may induce serious variable stresses in the piping. Some or all of these conditions in varying degrees of severity will exist in any piping system.

A system of steam piping may be considered as a structure to absorb a certain strain induced principally by temperature change in the pipe. A structure subjected to a certain stress will, if this stress exceeds the yield strength of the metal, distort, buckle, and collapse. This series of events will in general characterize a static failure of ductile material. A piping system, however, is not subjected to this type of loading. Once the specified strain is absorbed, no further motion leading to collapse is possible. Repeated applications of strain, provided the resulting induced maximum stress is beyond the endurance limit for the metal, will lead to fatigue fracture.

It is difficult to visualize a service loading condition which would lead to piping failure with a single application of load. Professor Hovgaard in one of his papers shows a photograph of a 9½-inch pipe bend bent back upon itself so that the angle between the two

* "A Manual for the Design of Piping for Flexibility by the Use of Graphs," by E. A. Wert, S. Smith and E. T. Cope, published by The Detroit Edison Company in 1934.

parts was about 30 degrees.* There was complete collapse of the cross-section but no sign of fracture anywhere.

Under service conditions the ultimate failure of piping will be fracture resulting from repeated applications of stress caused by temperature changes, pressure changes, vibrations, or from relative movement of pipe anchorages. Elastic failure signalized by the passing of the yield point has no direct connection with the type of failure to be expected in piping systems. It is clear that variable stress, rather than static stress, considerations are the proper basis upon which to investigate the strength of pipe.

13. *The Mechanism of Fatigue Failure.*—If a reasonably ductile material, such as the steel from which steam pipes are commonly made, is subjected to steady load or only a few cycles of repeated load, highly localized stress will not cause appreciable damage provided the average stress is below the yield point of the material. In the areas where high local stresses exist, yielding occurs and the stress tends to equalize across the section. Where the material is under many cycles of repeated load of sufficient magnitude, the localized yielding leads to embrittling in these areas from cold work. The sliding of layers of metal within crystals, one upon the other, gradually leads to the development of minute cracks which spread and enlarge, leading to ultimate fracture. It will be appreciated that the area of highly localized stress may be small and that plastic flow may occur in this region long before the elastic properties of the piece as a whole may appear to be effected.

It appears that for all metals a limiting stress exists below which no fracture will occur even after the application of an indefinitely large number of cycles of stress. This limiting stress for most steels is established between 10^6 and 10^7 cycles of reversed stress. As the stress is raised above this limit the failure will occur at fewer numbers of cycles. The limiting stress is known as the endurance limit of the metal. In all cases the endurance limit is well below the static tensile strength and has been determined usually to be a certain percentage of this latter value. There does not appear to be any connection between the endurance limit and the elastic limit as measured in static tests.

* "Further Research on Pipe Bends," p. 255, loc. cit.

Local areas of high stress may be caused by abrupt changes in sectional dimensions such as at a hole in a plate, or at a shoulder on a bar. Likewise scratches, grooves, notches, or internal discontinuities in a metal will cause localized concentration of stress. A serious condition arises when corrosion of metal is accompanied by cyclic stressing. The cyclic stress accelerates the corrosion and the corrosion in turn causes areas of high stress concentration in the corrosion pits and cracks. This mutually intensifying action will lead to early failure or failure at considerably lower stresses than indicated by values of the endurance limit.*

14. *The Limitations of the Theory of Elasticity in Analyzing Stress in Corrugated and Creased Bend Piping.*—The mathematical theory of elasticity based on the work of Saint Venant affords a means of determining the distribution of stresses in many complex shapes. Often, however, the application leads to differential equations which are impossible of solution. The fundamental assumptions underlying the theory are that the material in which the state of stress is to be investigated is perfectly elastic, that it is homogeneous, that it is infinitely divisible, and that it is isotropic. Although obviously these assumptions are not valid in the case of most materials, such as steel for instance, the method yields results which represent an average correctness for the aggregate of crystals which go to make up the structure of the metal. In certain cases where two-dimensional problems of elasticity are to be solved, recourse may be had to photoelastic methods where exact mathematical solutions may not be obtained.

The analysis of the state of stress existing in the corrugated and creased bend piping under various types of loading appears to be a problem for which no rigorous mathematical solution is possible.

15. *The General Scheme of Investigation.*—The presence of corrugations or creases in a pipe, while increasing the flexibility, also acts to increase the stress. If fatigue tests are made on these pipes so as to determine an endurance limit for each given type, it will develop that the stress at this endurance limit when computed from the ordinary Rankine formulas of mechanics is far different from

* For an excellent summary of this complex subject see "Corrosion-Fatigue of Metals," by H. J. Gough, the Journal of the Institute of Metals (England), Vol. XLIX, 1932. Also "The Influence of Chemically and Mechanically Formed Notches on Fatigue of Metals," by D. J. McAdam, Jr., and R. W. Clyne, Journal of the National Bureau of Standards, Vol. 13, October, 1934. This paper includes a selected bibliography.

the stress corresponding to the known endurance limit of the steel. For the purpose of determining the stress at which the pipes were operated in fatigue tests the bending formula was used

$$S = \frac{Mc}{I}$$

In this formula, M is the value of the applied bending moment, I/c is the value of the section modulus of the basic pipe. In order to determine this value, the moment of inertia, I, of the cross-section was computed from dimensions of the pipe before being corrugated or creased. The value of c was likewise taken as the radius of the outer surface of the basic smooth-walled pipe. This equation then yields values for stress, S, which are nominal values. The term "nominal stress" has been applied to this quantity. The ratio of the endurance limit for the steel to the endurance limit of the pipe, when the latter is expressed in terms of nominal stress, will then be a direct indication of the stress multiplication effect due to the corrugated and creased bend shapes. The writer has called this factor the "stress intensification factor" and has given it the designation N.

In addition to the determination of the stress intensification factor, a method was developed for determining the stress distribution in pipes from internal pressure and from applied bending moments. This method employs the use of strain gages applied to the outside of the pipe wall and involves the calculation of the critical stresses at the inner pipe wall surface derived from these external strain measurements.

IV. MATERIAL UNDER INVESTIGATION.

16. *Classification of the Various Types of Piping.*—This investigation covers three classes of pipe shapes, all constructed from 6-inch seamless steel tubing as follows:

Series A—Corrugated piping with nominal wall thickness 0.280 inches and shape of corrugation as shown on Fig. 1, Plate I.

Series B—Corrugated piping with nominal wall thickness 0.288 inches and shape of corrugation as shown on Fig. 2, Plate I.

Series C—Creased bend piping of a typical shape of crease as shown in Plate II and with nominal wall thickness 0.288 inches.

Complete tabulations of the pipe shapes investigated under each of the above classifications, together with details of heat-treatments and physical properties, are given in Plates III and IV.

17. *The Individuality of the Corrugation and Crease.*—In manufacturing a corrugated pipe the seamless steel tubing is placed in a press between a fixed and movable head. A gas-fired ring furnace is placed around the pipe and a narrow band heated. When the pipe has reached a proper temperature, it is compressed along its axis by the press, thus raising a corrugation in the heated section. When the compression has raised the corrugation a desired amount, the press is stopped, the corrugation quenched, and the furnace moved on to form the adjoining corrugation. The first motion of the press, in forming a corrugation, causes an upsetting of the metal in the heated band before the swelling starts. This action accounts for the increased wall thickness to be found in the radial section through the crest of the corrugation. If a bend is to be formed, the bending is done after the corrugating process. The fabrication is followed by a suitable annealing heat-treatment.

In forming a creased bend, the pipe is first filled with sand, locally heated at each crease in turn, and bent about 10 degrees at each crease. The creases thus bulge out from the original pipe wall and disappear at about two-thirds of the circumference of the pipe. This method of bending is said to have the advantage of permitting bending without a thinning of the wall on the outside of the bend and a thickening of the wall on the inside of the bend. Both corrugated and creased pipe may be made with much smaller radius bends than may be formed from corresponding smooth-walled tubing. The processes of manufacture are patented.

It is to be carefully noted that the method of manufacture results in each corrugation or crease being an individual with possibly different characteristics than its neighbors. Non-concentricity of the bore of the tubing with the outer surface may be serious if it provides a minimum section in the plane of maximum stress. Because of the localized stress conditions in these pipes, it is also highly important that no local defects such as inclusions, laps, seams, or surface cracks appear in the tubing, because of the danger that one of these areas of weakness occur in a location of maximum stress.

V. EXPERIMENTAL METHODS—THE DETERMINATION OF THE ENDURANCE LIMIT OF THE PIPING.

18. *The Types of Testing Machines Used.*—Two types of testing machines were used to determine the endurance properties of the pipes. One type of machine was designed to apply a specific stress to the specimen and the other type a specific strain.* The specific stress machine was, in principle, a large scale model of the rotating beam fatigue testing machine frequently used in the laboratory for determining the endurance limit of small specimens. The specific strain machines were designed to subject the pipe to alternating flexure by applying a constant deflection to the specimen. A total of four machines of both types was built.

A view of the specific stress rotating beam testing machine is given in Plate V. Naturally only the long tangent specimens could be adapted to this apparatus. As will be seen from the plate, the long tangents were mounted in trunnion bearings at either end. Babbitt metal journals cast on the pipe carried collars which supported the desired dead weight load. The pipes were rotated by a motor and belt drive through a universal joint. With equal loads applied to the arms hanging from each collar, the bending moment across the corrugated section of the tangent was very nearly uniform. Hence all corrugations between the journals were subjected to approximately equal stress, and failure could reasonably be expected to occur in the weakest corrugation. With the pipe section at rest under load, the stress distribution in the pipe wall may be considered to be symmetrical with respect to a horizontal plane through the longitudinal neutral axis of the pipe. Now if the pipe is rotated 180 degrees, the fibers which were in the top half-section under certain states of stress, will acquire an equal value of stress but with opposite sign. This same reasoning applies to the fibers originally in the bottom half-section. Thus for a complete revolution of the pipe each fiber passes through a complete stress cycle from a maximum positive value of stress through an equal minimum negative stress value back to the positive stress value. In this

* See "The Fatigue of Metals," by H. J. Gough (London), 1924. According to Gough's classification a specific stress testing machine is one which applies a definite load on the specimen for each cycle. The load remains constant throughout the test. A specific strain testing machine is one which produces a definite deformation of the specimen for each cycle. In this type the deformation remains constant throughout the test.

investigation the pipe rotative speeds ranged from 80 revolutions per minute to 220 revolutions per minute. The speeds were carefully selected so that the pipe rotated smoothly and without vibration.

Two kinds of specific strain machines were developed. Views of these machines may be seen in Plates VI and VII. Two machines of the first kind were built. These machines each consisted of a 25 horsepower motor driving, through suitable reduction gearing, a shaft at right angles to the armature shaft extended. This driven shaft carried at either end an adjustable eccentric upon which a journal was mounted. The eccentrics could be adjusted to give the desired crank throw for the journal. This journal was connected through a vertical connecting rod to a suitable point to apply load on the pipe section. The cycle frequency in these machines ranged from 30 to 90 cycles per minute.

The remaining specific strain type of machine was developed from the water end of a reciprocating pump. The pump was mounted with its rod horizontal and with a hydraulic system, under pressure, connected to either end of the pump cylinder. The water was alternately admitted to one end of the cylinder and discharged from the other through an arrangement of plug cocks whose action was governed by a linkage operated from an electric motor. With the apparatus in operation, the pump rod would move back and forth at a cycle frequency of approximately 20 per minute. The amount of rod movement could be adjusted by suitable setting of the linkage mechanism to the plug cocks and could be positively limited by adjustable stops attached to one of the pump tie rods. The outer end of the pump rod was attached to a pipe flange for endurance testing purposes. This machine was used for testing certain expansion U-bends only.

19. *The Correlation of Results Obtained from the Specific Stress and Specific Strain Fatigue Testing Machines.*—In the specific stress type of fatigue testing machine the bending moment could be determined from the applied loading. The machine was equipped with an electrical device arranged so that if the pipe under test should begin to rotate eccentrically or should fracture completely, the drive motor would stop. The time of actual failure, that is, when a fatigue crack developed completely through the pipe wall, was anticipated by a short period of slightly eccentric rotation of

the pipe before this eccentricity became severe enough to signalize complete failure.

In the specific strain fatigue test, the load-deflection characteristics of the pipes were first determined. The points of application of load and the direction in which the loads were applied corresponded to the conditions of loading of the specimens in the fatigue testing machine. With the load-deflection relationship established, the bending moment and hence the nominal stress, as defined above, could be determined for a given deflection setting of the machine. The usual procedure in testing was first to select a desired value for the maximum nominal stress to be imposed in the pipe specimen, then to determine from the beam formula the load to be applied, and finally to set the fatigue testing machine to provide the corresponding deflection as established from the load-deflection curve for that specimen. As with the rotating beam specimens, the point of failure was taken to be when a fracture had occurred completely through the pipe wall. In order to assist in the detection of such a crack, the pipes were coated with white wash before testing. No automatic means for stopping the machine at the time of failure could be devised. Hence close inspection of the pipes on the running tests was necessary.

It will be appreciated that, if yielding occurs as a fatigue crack develops, the actual load to maintain a constant deflection will be reduced in the specific strain type of machine. This reduction in applied load causes a proportional decrease in the applied stress which might tend to increase the time required for ultimate failure. Laboratory testing experience indicates, however, that fatigue cracks progress rapidly to failure when once started, and that close agreement in results may be expected in testing with specific stress and specific strain type machines.

20. *The Effect of Range of Stress on Fatigue Strength.*—When a material is subjected to cyclical variations of stress the range of stress is defined as the algebraic difference between the maximum and minimum stresses of the cycle. The range ratio is equal to the minimum stress divided by the maximum stress. Thus for the case of a completely reversed stress, or what is usually termed alternating stress, the range ratio is —1.0. A range ratio of 0 would indicate that the stress varied from 0 to a maximum. The range ratio can never have a value greater than 1.

Several investigators have developed formulas and diagrams in order to define the relationship between stress range and fatigue strength. Gerber's parabola, Goodman's dynamic law, the formulas of Launhardt and Weyrauch, and the formula developed by J. B. Johnson are perhaps the best known of the earlier work on the subject. Much of this work did not check particularly well with later experimental determinations. The Goodman diagram for instance, implies that the endurance limit of a metal under alternating stress is one-third of the static ultimate tensile strength. This has been proven to be incorrect for many metals, particularly in the ferrous group. This diagram, however, further implies that the ratio between the endurance limit for a stress range from 0 to a maximum and the endurance limit for a complete reversal of stress is 1.5 to 1. This value was investigated by Professors Moore and Kommers, who concluded that it was a reasonably safe assumption for all metals.* They proposed a formula having the general form of the Johnson formula but not based on any assumed ratio of endurance limit to ultimate static tensile strength and retaining the Goodman ratio of 1.5.

$$S_{max} = \frac{3\ S_{-1}}{2 - r}$$

where r is the range ratio,

S_{max} is the maximum unit stress during a cycle, and

S_{-1} is the endurance limit for alternating stress.

In the writer's investigation only two stress cycles were used; one a complete reversal of stress, and the other with stress varying from 0 to a maximum. In order to reduce the results of these tests to an equivalent basis to enable plotting failures associated with each of the two stress cycles to the same scale and on the same fatigue graph, the values of stress at failure in the case of the stress cycle from 0 to a maximum were divided by 1.5. This follows from the formula of Moore and Kommers.

21. *The Relationship Between Cycle Frequency and Endurance Limit.*—The stress cycle frequency does not appear to have much, if any, effect on the fatigue resisting properties of metals. H. J. Gough summarizes the results of various investigators in this field,

* "The Fatigue of Metals," Chapter VII, by H. F. Moore and J. B. Kommers.

who all concluded that no speed effect exists.* The cycle frequencies investigated varied from 150 to 5000 cycles per minute. The number of cycles of stress applied to a metal is said to be the controlling factor and the frequency of stress application is not significant. The case is very different when corrosive conditions are present together with the application of stress. The stress and corrosion have a mutually intensifying effect, but, since corrosion may also proceed without simultaneously applied stressing, the total time of the action is of great importance. The effect of cycle frequency on corrosion-fatigue is complex and depends upon many variables. This problem has been thoroughly studied by Dr. D. J. McAdam, Jr., who investigated cases of variations of cyclic frequency of from 1 per day to 10,000 per minute.† It may be mentioned that the cases of the cycle frequency of 1 per day include a boiler explosion after approximately 13 years' service.

22. *The Endurance Limit for the Steel from Which the Piping Used in This Investigation Was Fabricated.*—It was not considered practicable to machine satisfactory fatigue test specimens from the sample pipes used in this investigation. It is not necessary, however, to determine the actual endurance limit of this steel by direct test. It is generally accepted that a metal's endurance limit under cycles of reversed flexural stress may be expressed as a definite percentage of its ultimate tensile strength. No such correlation has been discovered between the endurance limit and any other physical property of a metal. The ratio of the endurance limit to the ultimate tensile strength has been termed the endurance ratio by Dr. McAdam, and for most steels will have a value of ranging between 0.40 and 0.50. Plate VIII gives a fatigue graph for a carbon steel whose composition was similar to the steel used in the pipe. This steel had a tensile strength of 63,000 pounds per square inch and an endurance limit of 27,000 pounds per square inch. The value of endurance ratio is hence 0.43. This value of endurance ratio has been applied to the observed tensile strength of the pipe steel in order to determine the value of its endurance limit.

In connection with Plate VIII it may be stated that the three points of highest stress on the fatigue graph were determined at a

* See "The Fatigue of Metals," Chapter V, by H. J. Gough.

† "The Influence of Stress Range and Cycle Frequency on Corrosion," by D. J. McAdam, Jr., Proc. of A. S. T. M., 1930, Part II, Technical Papers.

specimen speed of 50 revolutions per minute, while the points for the remainder of the graph were determined at 1450 revolutions per minute. The point representing the highest stress on this graph was obtained with a computed bending stress on the specimen actually greater than the ultimate tensile strength.

VI. EXPERIMENTAL METHODS—THE DETERMINATION OF THE FLEXIBILITY OF THE PIPING.

23. *The Necessity for Establishing the Load-Deflection Characteristics.*—Since the specific strain type of fatigue testing machine imposes a definte deflection on the pipe, the bending moments and the consequent applied nominal stresses could be determined only from a knowledge of the relationship between the load and deflection of the specimen under consideration. In order to establish the load-deflection characteristics, each pipe shape was mounted with one end firmly secured to a heavy steel foundation plate and dead weight loads applied to the other end. Deflections were measured by means of micrometers with increments of applied load. Small balls, acting as gage points for the measurements, were attached to the pipe flanges and to battens which were fixed to the foundation. Plates IX, X, XI and XII give views of typical arrangements for the load-deflection determinations.

24. *The End Conditions.*—Consider a pipe bend with its end flanges firmly fixed and subjected to an increase of temperature. The expansion of the pipe line will cause the flanges to exert certain forces and moments on the anchorages which may be resolved into a force perpendicular to the flange, a force parallel to the flange in the plane of the bend, and a couple also in the plane of the bend. This is called a fixed end condition. If the two forces continue to act but the couple is removed, the flange is thus permitted to rotate, and the condition is known as free end.

The fixed end condition was simulated in load deflection testing as shown in Plate X. Here the restraining force parallel to the flange in the plane of the bend is clearly 0. It is known from mechanics that a force and a couple acting in the same plane may be replaced by a single force whose moment arm is equal to the value of the couple divided by this force. It will be seen from the photograph that this is accomplished by the method of loading shown.

Here, as the load is applied, the flange moves in parallel planes. This condition was checked on test by a spirit level mounted on the upper flange. The proper value for the moment arm was first computed. Any subsequent minor adjustment necessary was indicated by the level.

Plate XI shows the simulation of the free end condition. Here the restraining moment is absent and rotation of the flange is therefore permitted.

There is a slight discrepancy between the state of affairs governing a load-deflection or fatigue test and the conditions actually obtaining in service. In the test conditions the forces and couples are applied on the cold or unexpanded pipe and hence on a pipe shape slightly different, from a dimensional standpoint, from the same pipe under elevated temperatures in service. This difference, however, is so slight that it may be disregarded for all practical purposes. It may be remarked that pipes in service will probably operate at some condition intermediate between the free end and fixed end conditions, and that the nature of these end conditions will have a decided effect on the stresses produced in the pipe itself.

25. *Results of Load-Deflection Determinations.*—Plates XIII to XVII, inclusive, give graphs showing the load-deflection relationship for each of the pipe shapes in the A, B and C series of piping. At least one pipe shape of each type was tested for each of the A, B and C series of piping. The remaining bends were checked for agreement with the typical similar shape before fatigue testing. The graphs also indicate the methods of loading and the points of deflection measurement. The effect of cold work on raising of the primitive elastic limit may be noted on those graphs which show the load-deflection characteristics after repeated loading.

26. *The Flexibility Factor.*—The flexibility factor V is defined as the ratio between the deflection of a section of corrugated or creased piping under a certain load to the deflection of a section of smooth-walled piping of the same length loaded in the same manner and by the same amount, both pipes considered to be the same basic wall thickness and same basic internal diameter. In brief, it is the ratio of the flexibility of the corrugated or creased pipe to the flexibility of a similarly dimensioned smooth-walled pipe. As is the case with the factor K, developed by Dr. Karman, the flexibility

factor V may be considered as modifying the flexural rigidity of the pipe shape considered.

In determining the flexibility of the piping, the grapho-analytical method of calculating deflections, combined with the principle of super-position, is used.* This method has been developed by The Detroit-Edison Company, which has published formulas for the areas of the bending moment diagrams, together with formulas for rotations and deflections, for various combinations of loadings applied to straight tangents and also quarter bends.†

It is proposed in the subsequent sections to investigate the value of the flexibility factor V for the following pipe shapes:

(*a*) The straight tangent, Series A.
(*b*) The straight tangent, Series B.
(*c*) The corrugated arc, Series A.
(*d*) The creased arc, Series C.

The flexibility factors to be developed will apply directly to the corrugated or creased bend shapes considered, and are entirely unassociated with the effect of added tangent lengths of piping. Thus, in computing the flexibility of corrugated tangents, the factor given applies to the corrugated section only, the effect due to the smooth pipe tangent length at the ends being eliminated by computation. Similarly, the flexibility factors given for the corrugated and creased arcs apply directly to the arc itself, and do not include the effect of tangent lengths beyond the 90 degrees of arc.

27. *The Flexibility of Straight Tangents–Series A.*—The moment area diagram for piece AC is shown as Figure 1, Plate XVIII.

(1) $EI\delta = \frac{\psi \text{ i}}{2} \frac{(4450\ V + 608) \times 75.75}{2}$ Case II.‡

(2) $I = 28.14$ (on basic pipe dimensions of 6.625 inches O.D. and 0.280 inches wall thickness).

(3) $\delta = 1.33 \times 10^3$ inches per pound load (Figure 1, Plate XIII).

* See "Applied Elasticity," by S. Timoshenko and J. M. Lessells, Section 23.

†A typical calculation involving the use of these methods is given in Appendix I.

‡ The case numbers mentioned in the calculations of Sections 27 and 28, refer to formulas shown in Plates XXXIX and XL. The methods used are demonstrated in Appendix I.

(4) $E = 29.7 \times 10^6$ pounds per square inch (by test).

Substituting from equations (2), (3) and (4) in (1), and solving for V:

(5) $V = 6.46$.

A similar computation for piece AA gives a value of $V = 6.4$.

28. *The Flexibility of Corrugated Arcs–Series A.*—The moment-area diagram for piece AH is shown as Figure 2, Plate XVIII. It is assumed that, as demonstrated in the previous section, the flexibility factor of the tangent sections of this bend is 6.5.

$$EI\delta_c = \Psi_b 1 = 372{,}000\ F \quad \text{down Case I}$$
$$= \Psi_f .0.7854\ R = 7275\ VF \quad \text{down Case V}$$
$$= \Psi_m .0.637\ R = 19{,}750\ VF \quad \text{down Case VII}$$
$$EI\delta_d = \Psi_c 1 = 754{,}000\ F + 81{,}600\ VF \quad \text{down Case I}$$
$$= \Psi_f \frac{2}{3} 1 = 160{,}100\ F \quad \text{down Case II}$$
$$= \Psi_m \frac{1}{2} = 23{,}600\ F \quad \text{down Case III}$$

(1) $\varepsilon EI\delta = 1{,}309{,}700\ F + 108{,}625\ VF$

(2) $E = 29.7 \times 10^6$ (by test)

(3) $I = 28.14$ (from section 27)

(4) $\delta = 2.50 \times 10^{-3} F$ (Figure 2, Plate XIII).

Substituting from (2), (3) and (4) in (1), and solving for V:

(5) $V = 7.2$

This factor is approximately the same as the value of the flexibility factor for the straight tangent. As might be expected it is some what higher, possibly because of the deformation of the corrugations where the pipe is formed into an arc.

Dr. Karman's K value for a smooth-walled pipe arc of similar major dimensions is as follows:

$$K = \frac{10 + 12\lambda^2}{1 + 12\lambda^2}$$

$$\text{where } \lambda = \frac{tR}{r^2} = 0.584 \text{ and } \lambda^2 = 0.341.$$

(6) hence $K = 2.76$.

This indicates that this theoretical smooth pipe arc would be 2.76 times as flexible as an arc where no flattening occurred. This value of K for the smooth arc is comparable to the value of V for the corrugated arc. Hence, V/K = relative flexibility = 2.61. Therefore the corrugated arc is 2.61 times as flexible as a theoretical smooth arc having the same major dimensions.

29. *The Flexibility of Straight Tangents–Series B.*—The flexibility of the corrugated tangents of Series B, computed in the same manner as for Series A, gives the following results:

Piece BA, V = 6.4
Piece BB, V = 6.6
Piece BC, V = 6.4.

30. *The Flexibility of the Creased Arc–Series C.*—The flexibility of the creased arc of Series C, computed in the same manner as the corrugated arc of Series A, gives the following results:

Piece CA, V = 5.1
Piece CB, V = 5.0
Piece CC, V = 5.0.

The corresponding value of K for a theoretical smooth pipe arc of the same major dimensions is 3.79. Hence, assuming an average value of V to be 5.0, the creased arc appears to be approximately 32 per cent more flexible than the theoretical smooth arc.

31. *A Summation of Flexibility Results.*—The flexibility factor for all tangents tested may be taken as 6. Although the corrugated arc may show a slightly higher flexibility factor, this value of 6 should also be used for the arc in conservative design. Because of the increased flexibility of a smooth arc due to flattening, the corrugated arc shows an advantage in flexibility over the smooth arc of 2.6 to 1. Likewise the creased arc has a flexibility advantage of about 1.3 to 1.

VII. THE RESULTS OF THE FATIGUE TESTS.

32. *The Method Used in Developing the Graphs of Endurance Properties.*—The results of the fatigue tests of each of the series of corrugated and creased piping are shown graphically in Plates XIX, XX and XXI. The scheme used in plotting the fatigue graphs is the usual one for presenting results of fatigue testing. The axis

of ordinates is scaled uniformly to represent the semi-range of alternating stress while the axis of abscissas is scaled logarithmically to represent the number of cycles. The points on the graph, in general, indicate failures of the designated pipe shapes. In some cases the applied nominal stress values were raised during the test. Where this was done paths are shown on the graph to show the manner in which the specimen was tested at increasing stress values. Where arrows alone are attached to points this does not represent failure but means that the specimen is to be considered as continuing to run beyond the plotted point. The bending moments and the nominal stress were computed at the point of actual fracture for each specimen. The method of calculation employed was similar to that shown in Appendix I for the single offset quarter-bend, piece AM. In cases where the stress cycle varied from 0 to a maximum, the range was transformed for plotting purposes, as previously explained, to an equivalent range for alternating stress. The plotting of the fatigue graphs follows the practice in use at the U. S. Naval Engineering Experiment Station. Weight was given to the individual points, giving due regard to the effects of understressing and over-stressing on those specimens whose applied stress was raised in the progress of the test.

In all cases the nominal stresses, as previously defined, were computed from the simple beam formula as the stresses which would be produced in a theoretical smooth-walled pipe of the same basic dimensions as the corrugated or creased pipe when subjected to the same applied forces. The longitudinal stresses only were computed and in the comparative theoretical smooth-walled bend no allowance was made for any increase of stress caused by flattening. The direct stresses resulting from the loading are a small percentage of the stress due to bending and have, therefore, been neglected as minor in computing the nominal stresses.

33. *The Testing of Individual Specimens.*—In order to show clearly the circumstances attending the failure of each specimen, the details of the testing will be stated, treating the specimens individually.

SERIES A—PLATE XIX.

As a basis for determining the nominal stress the comparative theoretical smooth-walled pipe was assumed to have an outside

diameter of 6.625 inches and a wall thickness of 0.280 inch, thus having a section modulus of 8.5. The nominal stress was taken to be equal to the applied bending moment at the plane of fracture divided by the section modulus.

Straight Tangents.

The straight tangents were all tested in the specific strain type machine. The loading was applied in the same manner as shown for the load-deflection determination in Figure 1 of Plate XIII. The connecting rod of the machine was applied to the end of an I-beam which was secured at right angles to the pipe axis at the upper flange.

Piece AA was tested at a deflection setting to give a nominal stress range of 10,540 pounds per square inch varying from 0 to a maximum. Failure occurred after 121,840 cycles in the crest of the second corrugation from the base of the pipe.

Piece AB was tested at a deflection setting to give a nominal stress range of 12,920 pounds per square inch varying from 0 to a maximum. Failure occurred after 38,100 cycles in the crest of the second corrugation from the base of the pipe.

Piece AC was tested at a deflection setting to give an alternating nominal stress of 6900 pounds per square inch. Failure occurred after 198,164 cycles in the crest of the sixth corrugation from the base of the pipe.*

Expansion U-Bends.

Piece AD was erected for test as shown in Plate XXII. This was a fixed end test and, in addition, the pipe was placed under 400 pounds per square inch hydrostatic pressure. In order to eliminate corrosion as a factor, a 2 per cent solution of potassium dichromate was used as the liquid. This solution has strong properties as a corrosion inhibitor. The pipe was first put under internal pressure and then deflected so as to give a nominal stress range of 10,100 pounds per square inch varying from 0 to a maximum. The load-deflection relationship for this pipe was found to be the same

* Both the corrugation and the crease protrude beyond the original pipe wall. The crest of a corrugation or crease is considered as the region of maximum pipe diameter. The trough of a corrugation or crease is considered as lying between two adjacent crests and is therefore in the region of basic pipe diameter. These definitions are to be considered as applying where the terms "crest" and "trough" are used in this paper.

with the internal pressure of 400 pounds per square inch as without this pressure. Failure occurred at the point of maximum bending moment on the crest of the first corrugation from one end. Although the stress due to the applied bending moment varied from 0 to a maximum, the internal pipe pressure produced a state of steady stress in the pipe which caused the total stress to vary from a stress value different from 0. For this reason, the failure of this particular specimen should not properly be plotted on the graph of Plate XIX. It has, however, been placed on the graph for comparison purposes but has been disregarded in determining the fatigue curve. For purposes of plotting, it was assumed that the stress range ratio was 0.

Piece AE was tested in the pump gear specific strain type machine in the free end condition. A deflection was selected to give a nominal stress of 7040 pounds per square inch varying from 0 to a maximum. Failure occurred in a crest of a corrugation at the point of maximum bending moment after 316,900 cycles.

Piece AF was tested in the pump gear machine with one end fixed and the other end guided. An alternating nominal stress of 6280 pounds per square inch was imposed on the pipe. Failure occurred after 75,200 cycles in a corrugation trough near the point of maximum moment. This point of failure was on the inner arc of the pipe bend and because of the high arc curvature the corrugations were crimped closely together. This was the only failure observed in a corrugated pipe which did not occur in the crest of a corrugation.

Quarter Bends.

Piece AG was tested in the specific strain type machine with an alternating nominal stress of 5160 pounds per square inch. Failure occurred in the crest of the twelfth corrugation from the base after 288,700 cycles.

Piece AH was tested in a similar manner and failed in the crest of the sixteenth corrugation from the base after 267,400 cycles with an alternating nominal stress of 4140 pounds per square inch.

Straight Tangent Rotating Beam Specimens.

Piece AJ failed in the first and second corrugation crests from one journal after 1,178,780 cycles at an alternating nominal stress of 3660 pounds per square inch.

Piece AK failed in the crest of the sixth corrugation from one journal after 136,020 cycles at an alternating nominal stress of 6000 pounds per square inch.

Piece AL failed after 1,475,370 cycles in the crest of the fifth corrugation from one journal with an alternating nominal stress of 3060 pounds per square inch.

Single Offset Quarter Bend.

Piece AM was tested in the specific strain type machine as shown in Plate XXIII. This was a special type of bend curved to a sharp radius so that the corrugations were considerably flattened along the outer axis of the bend. This piece was tested with a nominal stress of 8400 pounds per square inch at the point of failure, varying from 0 to a maximum. The failure occurred after 78,105 cycles at a point where the actual computed nominal stress was approximately 16 per cent less than exerted at the point of maximum computed stress. This specimen should not properly be plotted in Plate XIX but has been placed there for comparison purposes and was disregarded in determining the fatigue graph.

SERIES B—PLATE XX.

As a basis for determining the nominal stress the comparative theoretical smooth-walled pipe was assumed to have an outside diameter of 6.625 inches and a wall thickness of 0.288 inch, thus having a section modulus of 8.7. The nominal stress values, computed for the point of actual fracture, were derived as previously explained.

Straight Tangents.

All straight tangents were tested in the specific strain type machine in a manner similar to that described for the straight tangents of Series A.

Piece BA was tested with a nominal stress cycle of 10,500 pounds per square inch varying from 0 to a maximum and failed in the crest of the seventh corrugation from the top after 200,480 cycles.

Piece BB was tested with an alternating nominal stress of 7840 pounds per square inch and failed after 94,400 cycles in the seventh corrugation from the base of the pipe.

Piece BC was tested with an alternating nominal stress of 11,300 pounds per square inch and failed after 26,700 cycles in two places, at the crests of the fourth and sixteenth corrugations from the base, and on opposite sides of the pipe.

Quarter Bends.

Three quarter bends, pieces BD, BE and BF, were constructed with a creased arc and with one corrugated tangent leg. Thus for purposes of plotting on the fatigue graphs they were considered as belonging to both the B and the C series.

Piece BD was one of the first specimens to be tested. No information was available concerning the location of the fatigue curve in the stress-cycle plane. This specimen was therefore tested at increasing values of stress in order to determine as quickly as practicable some probable limits for the location of this curve. This specimen was tested first with an alternating nominal stress of 2880 pounds per square inch for 63,605 cycles. The stress was then raised to 6950 pounds per square inch for 100,058 additional cycles. In the next step the nominal stress was raised to 10,100 pounds per square inch and failure occurred after an additional 5630 cycles of alternating stress. The failure occurred in the trough of the lowest crease. The specimen is therefore plotted as continuing on the graph for the B series corrugated pipe, since no failure occurred in the corrugated leg, and as a failure on the graph for the C series creased pipe.

Piece BE was tested in the specific strain machine with an alternating stress of 2220 pounds per square inch. In addition to this loading, a steam pressure of 100 pounds per square inch was placed on the pipe. No failure occurred after 2,830,624 cycles. This specimen is therefore plotted as continuing on each of the B series and C series graphs. The steam pressure was disregarded in computing the range ratio and the nominal stress, and the point of discontinuing the test plotted on the fatigue graph for both the B and C series.

Following this test, the pipe was carefully inspected both externally and internally. No evidence of damage could be discovered. Since the test had been run with a value of stress somewhat lower than the endurance limit it was unlikely that any damage would

have occurred. This specimen was then erected for test as before but with 400 pounds per square inch internal hydrostatic pressure. A solution of potassium dichromate was used as a liquid. The deflection was adjusted to give an alternating nominal stress of 5740 pounds per square inch. A failure resulted in the trough of the lowest crease of the arc after 201,361 cycles. The crack was welded and the test continued. A failure resulted in the crest of the tenth corrugation from the base after an additional 11,200 cycles. These failures of piece BE with 400 pounds per square inch internal pressure have not been plotted on the fatigue graphs. It may be seen, however, by reference to the curves that these points of failure lie in close proximity to the curves.

Piece BF was subjected to a nominal stress range of 15,850 pounds per square inch varying from 0 to a maximum. It failed after 24,805 cycles in the fourth corrugation from the top. It was therefore plotted as a failure on the B series graph and as continuing on the C series graph.

Straight Tangent Rotating Beam Specimens.

Piece BG was tested in the rotating beam machine with an alternating nominal stress cycle of 4390 pounds per square inch and failed after 307,250 cycles in the crest of the third corrugation from one journal.

Piece BH was similarly tested and failed in the crest of the fifth corrugation from one journal after 328,200 cycles of 5040 pounds per square inch alternating nominal stress.

Piece BJ was similarly tested and ran for 6,205,860 cycles at an alternating nominal stress of 2240 pounds per square inch. The stress was raised to give a cycle of 3420 pounds per square inch and failure then occurred in the crest of the first corrugation from one journal after an additional 22,970 cycles.

SERIES C—PLATE XXI.

As a basis for computing the nominal stress the comparative theoretical smooth-walled pipe was assumed to have 6.625 inches outside diameter and a wall thickness of 0.288 inch. Nominal stresses at the point of actual failure were computed as explained previously.

Piece CA was among the first specimens tested. In order to determine probable limits for the fatigue curve in the stress-cycle plane this specimen was tested at increasing values of nominal stress. It was first tested at 2330 pounds per square inch alternating nominal stress for 15,600 cycles, then at 4630 pounds per square inch for an additional 35,600 cycles, then at 6820 pounds per square inch for an additional 100,000 cycles, and finally at 9270 pounds per square inch when failure occurred after an additional 43,200 cycles. Failure occurred in the trough of the lowest crease of the arc.

Piece CB was subjected to an alternating stress of 6900 pounds per square inch and failed, after 108,000 cycles, in the trough of the first crease from the vertical tangent.

Piece CC was subjected to an alternating nominal stress of 3260 pounds per square inch for 5,200,000 cycles. The stress was raised to 4110 pounds per square inch for an additional 2,100,000 cycles, and failure finally occurred after an additional run of 720,000 cycles at a nominal stress of 5120 pounds per square inch. Failure occurred in the trough of the second crease of the arc from the vertical tangent.

Expansion U-Bends.

Piece CD was tested similarly to piece AD but without hydrostatic pressure, as shown in Plate XXIV. A nominal stress of 11,650 pounds per square inch varying from 0 to a maximum was applied. It failed in numerous places after 260,000 cycles.

Piece CE was tested in the pump gear machine with one end fixed and the other end guided. An alternating nominal stress of 6720 pounds per square inch was applied and failure occurred in the trough of a crease approximately at the point of maximum moment after 260,000 cycles.

Piece CF was tested with fixed ends and with 400 pounds per square inch hydrostatic pressure. The testing arrangement was similar to that shown in Plate XXII for piece AD. The nominal stress due to the applied bending moment varied from 0 to 8380 pounds per square inch. This is equivalent for plotting purposes to an alternating stress of 5580 pounds per square inch. Because of the steady stress due to internal pressure the true stress varied above and below some value different from 0. The test of this specimen

was, therefore, not plotted in Plate XXI. The testing was discontinued after no failure had been observed following 1,040,000 cycles of applied stress. It may be seen, by referring this performance to the fatigue curve of Plate XXI, that the strength of this pipe was apparently not adversely affected by the internal hydrostatic pressure.

VIII. THE NATURE OF FATIGUE FAILURES.

34. *The Location of the Fatigue Cracks in Corrugated Piping.*—Plates XXV and XXVI show photographs of typical failures in corrugated piping. The failure shown in Plate XXV occurred with the rotating beam type testing machine, while the other shown on Plate XXVI occurred with the specific strain type machine. The appearance of the fractured surface which is so characteristic of fatigue fractures clearly indicates that the crack progressed from the inner surface to the outer surface of the pipe wall. This conclusion was checked in many cases by observing the progress of the cracks in axial planes across the plane of failure. Planes could be found where the crack could plainly be seen to have failed to work its way entirely through the pipe wall. A photomicrograph of a typical crack is shown in Plate XXVII. This specimen was removed from piece AM, the single offset quarter bend, from the corrugation just above the one where complete fracture occurred. The crack shown in this photomicrograph had started from the inside surface of the pipe wall but had not progressed entirely through the wall thickness.

With but one exception, all failures of the corrugated piping took place in the transverse plane through the crest of a corrugation. From a study of the probable distribution of stresses in the pipe wall it seems clear that this would be the location of highest stress, and that the critical zone would be at the crest of a corrugation on the inside of the pipe, which is, of course, normally the surface of sharpest curvature. The one exception concerned the failure of piece AF, which has already been noted. Here failure occurred in the trough between two corrugations and the crack appeared to originate from the outside surface of the pipe. In this case the corrugations had been strongly crimped together on the inner side of the arc of the bend and the longitudinal radius of curvature of the surface at the point of failure appeared to be at least equally as

small as the radius of curvature of the inner surface of the adjoining corrugations.

Because, in general, the fatigue cracks do not start from the outside, it may be expected that a failure of corrugated pipe in service will not be detected until the crack has progressed entirely through the metal. The first warning in such a case may be a steam leak or perhaps a complete rupture. Once started, the progress of a fatigue crack is very rapid and a complete traverse of the metal will occur relatively soon after the inception of the crack.

35. *The Location of Fatigue Cracks in Creased Piping.*—In all cases of failure in creased piping the crack appeared to start from the outer surface and proceed inward. Without exception the failure occurred in the trough between two creases. This is the reverse of the manner in which failures occurred in the corrugated piping where the crack advanced from the inner to the outer surface.

The case of specimen CD, an expansion U-bend, presented an interesting example of fatigue failure. As has been explained, this specimen was tested with fixed ends with a stress cycle varying from 0 to a maximum. Hence the creased arcs were all on the compressive sides of the bends. The ultimate failure occurred in a zone of maximum bending moment from a crack developing from the outer surface inward and located in a trough between two creases. Longitudinal strips were cut from the pipe in the plane of the pipe bend. One strip was taken from half of each creased arc. The half chosen was the one subjected to the maximum bending moment. All four strips thus chosen were theoretically equally stressed. Plate II shows a photograph of the four strips selected for examination. The inner pipe surface is on the right of each strip. A total of 16 creased crests and 12 troughs was provided in these strips. A microscopic examination showed that 7 troughs contained incipient cracks, while 6 crests also showed cracking. There was a total of 10 cracks in the crests and 12 cracks in the troughs.

The appearance of the large number of cracks in the specimen CD is probably due to the nature of the test. It is possible that under the compressive cycle the crack associated with ultimate failure would not progress from inception to fracture with sufficient rapidity to fail to provide time for additional cracks to develop.

The creased crests, however, were probably in a state of tensile stress which might aid the rapid propagation of a fatigue crack.

In the case of a creased bend it appears probable that the appearance of an external surface crack would precede failure. However, the progress of the crack is likely to be so rapid that routine inspections could hardly be depended upon to predict such failure.

IX. THE ANALYSIS OF STRAIN.

36. *The Purpose of the Analysis.*—The results to be sought in this analysis are the expressions defining the state of strain at the inner surface of the pipe, more particularly, at the point of fatigue failure. In order to obtain these expressions the state of strain at the outer surface is determined experimentally and with certain assumptions as to the distribution of strains through the wall section, the state of strain at the inner surface will be deduced. The method is here presented as applying to a corrugated pipe. As will be shown, the same method, with some simplification, may be applied to the determination of the strains in the critical zone of probable failure in a creased pipe. The ultimate object in determining the strains is to establish the state of stress. The method of interpreting the results of the strain analysis in terms of stress will be developed in later sections.*

37. *Fundamental Considerations.*—In the following analysis certain fundamental assumptions are made. It is assumed that the material is perfectly elastic, that transverse cross-sections in bending originally plane remain so, and that the principle of superposition holds. The term "strain" will refer to the value of this quantity per unit length. Where the integrated value of strain over a length is to be indicated, the term "total strain" will be used.

A corrugated pipe shape may be considered to be formed by the revolution of the curve of the neutral axis of the corrugated wave shape about a longitudinal axis. In the zone of greatest interest, namely, the crest of a corrugation, the radius of curvature of the surface thus formed is large with respect to the thickness of the pipe wall. The radius of curvature of the corrugation itself is, however, not large with respect to the wall thickness. These two

* The writer has benefited by being permitted to refer to Dr. W. M. Coates' unpublished monograph on stress distribution in shells.

radii are designated as R_2 and R_1, respectively, and are indicated in Figure 5 of Plate XXIX. This diagram shows a longitudinal section and a hoop section of a corrugation. The system of designating the axes is also shown in the diagram. Thus 1 refers to the longitudinal direction, 2 to the hoop direction, and 3 to the radial direction. These three axes are mutually perpendicular.

The fact that R_1 is small compared to the wall thickness causes a thin shell analysis of strain to be invalid and necessitates knowledge of the position of the longitudinal neutral surface with respect to the inner and outer surfaces, as well as knowledge of the manner of strain distribution through the section, in order to deduce the value of inner surface strain from outer surface measurements.

The strains across the shell thickness are considered as of two classes:

(*a*) Uniformly distributed or membrane strains,
(*b*) Non-uniformly distributed or bending strains.

These categories include strains in the shell caused directly from the shell loading or those induced in one direction by strain in a direction perpendicular to it. The bending strains are associated with a relative change in angular displacement of the normals to the neutral surface.

Compressive strains are to be considered as negative in sign and tensile strains as positive.

38. *The Strain Distribution.*—

Notation.

E_{1_e} = Resultant longitudinal strain at outer surface of shell.
E_{2_e} = Resultant hoop strain at outer surface of shell.
ε_{1_e} = Longitudinal bending strain at outer surface of shell.
ε_{2_e} = Hoop bending strain at outer surface of shell.
ε_{1_m} = Longitudinal membrane strain constant throughout the shell thickness.
ε_{2_m} = Hoop membrane strain constant throughout the shell thickness.

The subscript i, used in place of e, refers to the inner shell surface.

The linear strain measured at the shell surface will consist of two components, bending strain and membrane strain. If the bending strain may be determined from the measurement of angular normal displacements, and if the total strain may be determined from measurements of surface extension, the value of the membrane strain may be obtained by subtraction. These considerations lead to the equations for the resultant strains—

(1) $E_{1_e} = \varepsilon_{1_e} + \varepsilon_{1_m}$

(2) $E_{2_e} = \varepsilon_{2_e} + \varepsilon_{2_m}$

These equations define the longitudinal and the hoop strains at the outer surface of the shell. At the inner surface, the bending strains will have reversed sign and will, in general, differ in absolute value. The membrane strains at the inner surface will, by definition, be identical in size and value with the membrane strains at the outer surface. Assume that

$$\left|\frac{\varepsilon_{1i}}{\varepsilon_{1e}}\right| = k_1 \text{ and that } \left|\frac{\varepsilon_{2i}}{\varepsilon_{2e}}\right| = k_2$$

then—

(3) $E_{1_i} = E_{1_e} - \varepsilon_{1_e} - k_1\varepsilon_{1_e}$

(4) $E_{2_i} = E_{2_e} - \varepsilon_{2_e} - k_2\varepsilon_{2_e}$

These equations define the longitudinal and hoop strains at the inner surface of the shell.

In the hoop section, the radius of curvature of the neutral surface is large compared to the shell thickness. Hence the position of the neutral surface may be taken as identical with the mid-surface as with thin shells. The value of k_2 is therefore one, and the bending strain is distributed linearly across the thickness.

In the longitudinal section, the shell must be considered as thick and the value of k_1 must be mathematically or experimentally determined since the bending strain distribution across the section will not follow a linear law.

39. *The Bending Strain.*—

(5) $\varepsilon_z = \dfrac{z\,d\Theta}{(R + z)\,d\beta}$ (By definition, from Figure 1, Plate XXIX)

where ε_z is the bending strain on the surface whose radius of curvature is $(R + z)$, or—

$$(6)\quad \varepsilon_z = z\frac{d\Theta}{ds'}$$

where $ds' = (R + z)\, d\beta$

40. *Methods of Strain Measurement—The Resultant Strains.*—The Huggenberger Tensometer was used for measuring directly the resultant strains. A view of this instrument may be seen in Plate XXX. The movement of the knife edges is magnified approximately 1200 times on the scale. The knife edges may be adjusted to various gage lengths. In this investigation gage lengths of one-quarter inch in the longitudinal direction, and one inch in the hoop direction, were used.

41. *Methods of Strain Measurement—The Bending Strains.*—A method of measuring bending strains has been described by Huggenberger.* Plate XXXI shows a bending strain gage constructed after his schematic diagrams. The bases of the instrument legs are fixed to small rods soldered to the surface to be measured. Figure 2 of Plate XXIX gives a diagrammatic sketch of the action of the instrument in measuring bending strain. The dotted lines in the diagram indicate the relative positions of the instrument parts after the surface to which it is attached has been placed in pure bending. The angular rotation of the legs of the instrument may be considered as taking place about the neutral axis MN. From the Figure, it will be seen that—

$$(7)\quad RS = (d + h)\, d\Theta$$

$$(8)\quad dL = RS\left(\frac{m + n}{m}\right) = (RS)g$$

where g is the magnification factor of the instrument.

$$(9)\quad d\Theta = \frac{dL}{g\,(d + h)}\ [\text{from (7) and (8)}]$$

then

$$(10)\quad \varepsilon_e = \frac{h}{g\, ds'\,(d + h)}dL\ [\text{from (9) in (6) assuming } z = h]$$

* "Ueber die Festigkeit der Gewölbten Böden und der Zylinderschale by Höhn, Berlin, 1927, Part II, by Huggenberger.

In pure extension the reading of the strain gage will be simply the relative movement of the rods at the surface multiplied by the multiplication factor, or

$$(11)\quad dL' = gds'\varepsilon_m$$

If the surface on which the strain gage is applied is subjected to a combination of bending and tensile strains, then the reading of the instrument will be the sum of the individual readings of each component considered separately, or

$$(12)\quad dL'' = dL + dL'$$

then

$$(13)\quad dL'' = gds'\varepsilon_m + gds'\left(\frac{d + h}{h}\right)\varepsilon_e$$

Generalizing equation (1)—

$$(14)\quad E_e = \varepsilon_e + \varepsilon_m$$

Solving equations (13) and (14) as a simultaneous system—

$$(15)\quad \varepsilon_e = \frac{h}{d\ (ds')}\left(\frac{dL''}{g} - ds'E_e\right)$$

This equation may be written as applying to the longitudinal and hoop directions as follows:

$$(16)\quad \varepsilon_{1e} = \frac{h_1}{d\ (ds')}\left(\frac{dL''}{g} - ds'E_{1e}\right)$$

$$(17)\quad \varepsilon_{2e} = \frac{h_2}{d\ (ds')}\left(\frac{dL''}{g} - ds'E_{2e}\right)$$

These two equations, in addition to equations (1), (2), (3), and (4), are sufficient to determine the state of strain at the inner surface from measurements made on the outer surface. E_{1e} and E_{2e} are determined directly from the tensometer readings, knowing the multiplication factor of this instrument and the gage length. The value for dL″ is taken directly from the reading of the bending strain gage. The application of these formulas, however, involves the value of h, which is the distance from the outer surface to the neutral axis and also involves values of k_1 and k_2, which are the

absolute values of the ratios of the bending strains at the inner surface to the bending strains at the outer surface in the longitudinal and hoop directions, respectively. As has been previously explained, the value for k_2 may be taken as 1.

The value for h_2 may likewise be taken as equal to one-half the wall thickness. The determination of the proper values for h_1 and k_1, however, requires further analysis. The solution for stress (and strain) distribution in curved bars in pure bending is applied to this analysis.

42. *The Determination of the Position of the Neutral Axis and Hence the Values of h_1 and k_1.*—The location of the planes of failure in the fatigue tests indicates that the section of principal interest lies at the intersection of a transverse plane passing through the crest of a corrugation, with the pipe wall. Because of the initial upsetting of the metal when the corrugation is formed, the outer and inner surfaces of the corrugation in a longitudinal plane will not have the same centers of curvature. For purposes of analysis the outer radius of curvature at the critical section is taken to be equal to the radius of curvature of the inner surface plus the wall thickness at this section. It is further assumed that the corrugation is symmetrical about an axis defined by the transverse plane for a sufficient arc length to encompass the region over which the strain measurements were made. With these approximations and assumptions the solution for stress distribution in pure bending of curved bars is used for the determination of the position of the neutral axis of a longitudinal section of the pipe wall.* The radius of curvature of the hoop section is considered to be large in proportion to the radius of curvature of the longitudinal section and hence to exert no sensible effect on the position of the neutral axis of this latter section.

The value of the tangential stress in a curved bar in pure bending is given by the equation:—

$$(18)\quad \sigma_\Theta = -\frac{4M}{N}\left(\frac{-a^2b^2}{r^2}\log b/a + b^2\log\frac{r}{b} + a^2\log a/r + b^2 - a^2\right)$$

where M = bending moment, and

$N = (b^2 - a^2)^2 - 4a^2b^2(\log b/a)^2$, and

a, b and r are the radii of curvature of the inner surface, outer surface and neutral surface, respectively.

* The solution here given is from "Theory of Elasticity," Chapter 3, by S. Timoshenko.

The right hand member of equation (18) was set equal to 0 and solved for r/a in terms of b/a. The result of this solution is shown in Plate XXXII. From this graph, knowing the values of b and a, the value of r and hence the location of the neutral axis may readily be determined.

Equation (18) was also solved for simultaneous maximum and minimum values for various values of b/a. The result of the computations are given graphically on Plate XXXIII where the values of k_1 may be read directly from the ordinate scale. These values for k_1 are practically the same as would be obtained by assuming a hyperbolic distribution of the bending strain across the shell thickness. In the determination of the values of k_1 and h_1 in the piping under test, the values for the dimensions of the radii of curvature of the inner and outer corrugation surfaces were taken as the average of a large number of measurements of corrugated sections. The average results of these measurements are shown in Figure 4 of Plate XXIX.

X. THE ANALYSIS OF STRESS.

Notation.

$\sigma_1, \sigma_2, \sigma_3$ = Normal components of stress parallel to 1, 2 and 3 axis.

ν = Poisson's ratio.

E = Modulus of elasticity.

$\varepsilon_1, \varepsilon_2, \varepsilon_3$ = Strains in 1, 2 and 3 directions.

σ_n = The value of stress at the endurance limit in reversed flexural stress.

σ_e = An equivalent tensile stress.

43. *The Determination of the Principal Stresses.*—It may be seen from considerations of symmetry that the principal axes at the inner surface of the crest of a corrugation are as indicated in Figure 5, Plate XXIX. The values of strain along axes 1 and 2 may be determined as explained in the previous section on strain analysis. It will now be shown, given these strain values in the longitudinal and hoop direction, how the values of the principal stresses may be determined.

It follows from Hooke's law that the three normal components of stress parallel to three mutually perpendicular axes may be expressed:—*

$$(19)\ \sigma_1 = \frac{\nu E}{(1+\nu)(1-2\nu)} e + \frac{E}{1+\nu} \varepsilon_1$$

$$(20)\ \sigma_2 = \frac{\nu E}{(1+\nu)(1-2\nu)} e + \frac{E}{1+\nu} \varepsilon_2$$

$$(21)\ \sigma_3 = \frac{\nu E}{(1+\nu)(1-2\nu)} e + \frac{E}{1+\nu} \varepsilon_3$$

where—

$$(22)\ e = \varepsilon_1 + \varepsilon_2 + \varepsilon_3$$

If the axes selected are the principal axes, the shearing components of stress vanish and equations (19), (20), and (21) completely define the state of stress on the sides of an element which are parallel to the coordinate planes.

The methods of strain analysis yield values for ε_1 and ε_2 (which are equivalent to E_{1_1} and E_{2_1} determined as described above), thus considering equations (19), (20), (21), and (22) there are four equations and five unknowns. This system of equations is then insufficient to provide a solution for the unknown quantities. In the cases of pipe loadings from bending moments alone, the value σ_3 is clearly 0. Also where the loading is from internal pressure, the value of σ_3 is equal to the value of this pressure. Hence in all cases the value of σ_3 may properly be assumed and the system of equations solved for σ_1 and σ_2.

With the assumption of the value for σ_3, the values of the principal stresses are now known.

44. *The Maximum-Shear Theory Applied to Fatigue Failures.*—The endurance limit of metals in shear, according to data assembled by Moore and Kommers, has a value of about 55 per cent of the endurance limit in reversed direct stress.† This fact indicates that the maximum-shear theory of the failure of metals is a safe approximation for cases where fracture occurs from fatigue.‡

* See "Theory of Elasticity"—Timoshenko—Introduction—or any text on elasticity.
† "Fatigue of Metals," by Moore and Kommers.
‡ See paper by J. J. Guest, loc. cit.

Since the maximum shearing stress is equal to half the difference between the maximum and minimum principal stresses, the condition for failure from fatigue is:

$$1/2\ (\sigma_1 - \sigma_3) > 1/2\ \sigma_n$$

assuming $\sigma_1 > \sigma_2 > \sigma_3$, and in which σ_n is the endurance limit in reversed direct stress, or in reversed flexural stress, since the value of the endurance limit should be the same in either case. An equivalent tensile stress may be defined as—

$$(23)\quad \sigma_e = (\sigma_1 - \sigma_3)$$

and hence the maximum shear stress is $\frac{\sigma_e}{2}$.

XI. THE VALUES OF STRESSES DERIVED FROM STRAIN MEASUREMENTS ON CORRUGATED PIPE.

45. *The Results of Strain Measurements.*—In order to demonstrate the application of the methods of analysis developed in the preceding sections, strain measurements were made on two corrugations of the tangent section of a quarter bend. Piece BE was erected for this test with the corrugated tangent leg vertical and measurements were made on the crests of the fourth and sixth corrugations from the bottom of the section. These corrugations were designated by the letters N and I, respectively. Two individual methods of loading were employed to produce strain. In one method a bending moment was applied to the pipe with no internal pressure, and in the other internal hydrostatic pressure was applied with no external loading or restraint. In this latter method of loading, the pipe was provided with blank flanged ends.

On each corrugation eight stations for measurements were established. These stations were placed with equal circumferential spacing along the crest of the corrugation. It was necessary to measure the longitudinal strains and hoop strains both with the bending strain gage and the tensometer at each station for each method of loading.

The results of the strain readings are pictorially tabulated in Plates XXXIV to XXXVII, inclusive. The readings reported are the result of the application of 32,750 inch-pounds or 19,650 inch-pounds bending moment or the application of 600 pounds per

square inch internal pressure as indicated on the diagrams. These values for bending moment and internal pressure have no direct connection with the fatigue tests, they were simply convenient points at which to take measurements. By applying the loading in successive increments, it was determined in each case, that the reading was proportional to the loading beyond the values reported. These readings can therefore be divided by the applied bending moment or by the applied pressure, as the case may be, and yield values of instrument reading per inch-pound applied bending moment, or pounds per square inch applied internal pressure. This scheme was employed in carrying out the calculations so that the final results for stress may be multiplied by values for moments or internal pressure to obtain stress values for the selected conditions.

46. *The Results Obtained for Values of Principal Stresses and Equivalent Stresses.*—With the methods previously described the external strain measurements were used to derive the values of the principal stresses at each of the stations on each corrugation. In the case of loading due to bending moment, $\sigma_3 = 0$, and in the case of loading due to internal pressure, $\sigma_3 = 400$ pounds per square inch. The results of the calculations for the principal stresses are tabulated below. The location of the points of measurement with reference to the pipe axes is shown in a diagram on Plate XXXVIII.

TABULATION OF PRINCIPAL STRESSES DUE TO APPLICATION OF BENDING MOMENT ONLY.

Crest of Corrugation N.

Position	*Principal Stresses in pounds per square inch per 100 inch-pounds moment*		
	σ_1	σ_2	σ_3
1	—74.3	—10.1	0
2	—62.1	—8.9	
3	—0.6	0	
4	+35.8	+3.5	
5	+48.5	+3.2	
6	+45.5	+5.9	
7	—8.0	—0.8	
8	—52.3	—6.4	

Crest of Corrugation I.

Position	*Principal Stresses in pounds per square inch per 100 inch-pounds moment*		
	σ_1	σ_2	σ_3
1	—84.3	—13.6	0
2	—49.3	—5.8	
3	+0.2	+0.8	
4	+33.6	+4.0	
5	+68.5	+14.3	
6	+33.3	+3.1	
7	+1.3	+0.6	
8	—52.5	—7.4	

It may be seen that with one exception, and this in a region of low stress, the greatest principal stress is σ_1 and the least principal stress is σ_3 at each point of measurement. The value of σ_e will therefore, from equation (23), always be equal to the value of σ_1 minus 0, in the case of loading due to bending moment, and will be equal to σ_1 minus the value of the internal pressure in the case of loading due to internal pressure. In the region of highest stress, the value of the radial stress corresponding to pressures for which the pipe was designed is small compared to the value of σ_1 and may be neglected without appreciable error. Hence for all cases it may be assumed $\sigma_e = \sigma_1$. This indicates that the hoop stresses, being of intermediate value between the longitudinal and radial stresses, have no influence on the strength of the pipe.

The value of the equivalent stress at each point of measurement, when an external bending moment was applied, is shown in Plate XXXVIII. This stress is plotted in values of pounds per square inch per 100 inch-pounds moment. Thus to obtain the equivalent stress induced by a known moment, the value from the curve should be multiplied by the value of the known moment divided by 100.

The values of the equivalent stresses caused by internal pressure only are as follows:

Crest N, $\sigma_1 = 52.4$ pounds per square inch per pound per square inch internal pressure.

Crest N, $\sigma_2 = 15.1$ pounds per square inch per pound per square inch internal pressure.

Crest N, $\sigma_e = 51.4$ pounds per square inch per pound per square inch internal pressure.

Crest I, $\sigma_1 = 40.1$ pounds per square inch per pound per square inch internal pressure.

Crest I, $\sigma_2 = 11.8$ pounds per square inch per pound per square inch internal pressure.

Crest I, $\sigma_e = 39.1$ pounds per square inch per pound per square inch internal pressure.

Thus, at the designed working pressure of this pipe; namely, 400 pounds per square inch, the maximum stress observed was over 20,000 pounds per square inch. The significance of these values for stresses caused by internal pressure will be discussed later.

As in the case of stresses caused by bending moments the intermediate stress value, the hoop stress, has no influence on strength.

XII. THE STRESS INTENSIFICATION FACTOR.

47. *The Value of Stress at the Endurance Limit.*—The endurance limit is a physical property of a metal and has a definite value. In the case of the steel used in the manufacture of the piping under investigation, the endurance limits are as follows:

Series A—25,100 pounds per square inch.
Series B—24,600 pounds per square inch.
Series C—25,300 pounds per square inch.

These values were determined as 43 per cent of the average tensile strength of the specimens in each series of pipes.

In the asymptotic portion of the endurance graphs, the stress in the metal must be at the endurance limit. The stress intensification factor has been defined as the ratio between the endurance limit and the nominal stress at this point on the fatigue graph. The values of the stress intensification factors obtained in this manner are:

Series A, $N = 8.7$
Series B, $N = 9.5$
Series C, $N = 6.1$

These values indicate that the nominal stresses have been effectively multiplied at the endurance limit by the factors indicated. It is considered that the discrepancy between the Series A and Series B pipes may partially be explained by the relatively few points available to determine the endurance limits on the fatigue graphs.

48. *The Stress Intensification Factor from Computed Stresses.*—The maximum stress shown on Plate XXXVIII appears to be at point 8 on I corrugation and has a value of 84.3 pounds per square inch per 100 inch-pounds moment. At the endurance limit, as defined on Plate XX for Series B piping, the bending moment is 23,500 inch-pounds. Hence the value of $\sigma_1 \text{max.} = 19{,}800$ pounds per square inch. This value gives a value for stress intensification factor of 7.3. This compares with the value of 9.5 for the B series obtained from the endurance testing method.

These results are in good experimental agreement. The value of 9.5 obtained from the endurance tests is probably slightly high because it is derived from a value for endurance limit associated with tests of small, highly polished, specimens tested at high frequency. The actual steel in the pipe has a rough surface and if this surface were duplicated in a small specimen, the resulting endurance limit would undoubtedly be somewhat lower than for the highly polished specimen. The speed effect probably has no influence as such. However, if corrosion is present the rough pipe surface offers more opportunity for attack during the time of test than the polished one. The true value of the endurance limit of the metal itself as actually in the pipe is probably somewhat less than the value of 24,600 pounds per square inch. The strain measurements upon which the computed stress results are based were made over a finite gage length across a region where the strain was varying. The results, therefore, represent an average which is somewhat below the maximum. The value of the stress intensification factor computed from the stresses indicated from these measurements will probably be low. The measurements were made primarily to demonstrate the method of determining the stresses and not for the purpose of developing the stress intensification factor. The ultimate failure of this piece, BE, in fatigue test, did not occur in either of the corrugations in which the stress was measured. Examination of the pipe after the test, however, showed that the metal had begun to crack at the inner surface of the crest of one of the corrugations selected for strain measurement.

XIII. THE INFLUENCE OF INTERNAL PRESSURE.

49. *The Action of Internal Pressure in Pipes.*—It has been mentioned that variations in internal pressure may be a source of cyclic

stress. The high values of equivalent stress for moderate values of internal pressure indicate that severe stressing from this cause might occur. However, it should be noted that the highest principal stress, the longitudinal, was caused from the ends of the pipe being closed while pressure was applied. This would be equivalent to the pipe line in service with a valve in the line closed and linear expansion unrestrained.

If a pipe line carrying internal pressure is considered as having the ends open, the only stresses acting are the radial and hoop stresses. The longitudinal stresses may then be neglected. In this case, for moderate pressures, the value for σ_2 due to internal pressure added to the value for σ_2 due to bending moment would probably yield values which would still be intermediate between values of σ_1 and σ_3. This question should be investigated for each case considered. The value of σ_2 with an open end pipe under internal pressure will be somewhat different from the values obtained by experiment with a closed end pipe. If the open end condition can not be readily obtained in a test, the value of σ_2 due to internal pressure which will probably be low compared to the other stresses acting, may be computed with a sufficient degree of accuracy from the ordinary boiler formula

$$\frac{pD}{2t} = \sigma_2$$

where p = Internal pressure,
D = Diameter of pipe, and
t = Wall thickness.

It will be seen that there are many ways in which the stresses due to internal pressure may operate to affect the strength of piping. The pressure stresses may themselves be cyclic, in certain conditions they may be disregarded, in other conditions they may act as steady stresses upon which cyclic stresses due to other causes may be superimposed. This latter condition will operate to shift the position of the cyclic stress range but not its value.

In the fatigue tests of specimens AD, BE, and CF this condition was obtained. The pipe was placed under internal pressure and then the cyclic stress was imposed upon the steady stress.

50. *The Effect of Internal Pressure on Flexibility and Strength.*—Load-deflection tests made on pieces AD, BE and CF without internal pressure and with 400 pounds per square inch internal pressure showed practically no difference. The flexibility of these pipes was unaffected by the internal pressure.

These three pipe shapes were each fatigued tested with 400 pounds per square inch internal pressure. Four failures resulted from these tests. Piece AD, a corrugated U-bend, provided one failure. Piece BE, a quarter bend with creased arc and corrugated tangent leg, failed in the creased arc, was repaired by welding and later failed in the corrugated leg. Piece CF, a creased U-bend provided the fourth failure. In each case, disregarding the stresses from internal pressure and referring the failures to the proper fatigue graph for the applied nominal stresses induced by the bending moments, the points of failure lie in close proximity to the curves. These results indicate that the internal pressure had no appreciable effect on the endurance properties of the bends thus tested.

XIV. THE DETERMINATION OF LIMITING DESIGN STRESSES FOR CORRUGATED AND CREASED PIPING.

Notation.

σ_m = Constant mean stress
σ_r = Reversed stress
$\sigma_{y.p}$ = Yield Point in tension
σ_n = Endurance limit in reversed flexural stress
f = Factor of safety
N = Stress intensification factor

51. *The Method of Determining Safe Working Stresses.**—It is assumed that the piping in service will be subjected to variable stress. Any kind of direct variable stress may be resolved into a constant mean stress:

$$\sigma_m = \frac{\sigma_{max.} + \sigma_{min.}}{2}$$

and a reversed stress:

$$\sigma_r = \frac{\sigma_{max.} - \sigma_{min.}}{2}$$

* For a complete description of this method see paper by C. R. Soderberg, "Factor of Safety and Working Stress," APM-52-2 Transactions of Amer. Soc. Mech. Eng., 1930; also, "Strength of Materials," Vol. II, by Timoshenko.

From these equations it follows that—

$$(24)\quad \sigma_{max} = \sigma_m + \sigma_r$$

With σ_m as abscissa and σ_r as ordinate, a diagram such as shown in Figure 3, Plate XXIX, may be made on which any variable stress can be represented by a point in the σ_m, σ_r plane. In this diagram let A represent the yield point in a static tension test ($\sigma_r = 0$) and let B represent the endurance limit in reversed stress ($\sigma_m = 0$). It may then be assumed that the limiting stress conditions are represented by points along the line AB. If a factor of safety, f, is applied to OA and OB, points determining the line CD parallel to AB are obtained. This line represents safe stress conditions and from it may be determined safe values of σ_m and σ_r for any value of the ratio σ_r/σ_m. Referring to this Figure, the following equations result from purely geometrical considerations:

$$(25)\quad \frac{1}{f} = \frac{\sigma_r}{\sigma_n} + \frac{\sigma_m}{\sigma_{y\cdot p}}$$

$$(26)\quad \sigma_r = \frac{\sigma_n}{f} - \frac{1}{1 + \frac{\sigma_n}{\sigma_{y\cdot p}}\,\frac{\sigma_m}{\sigma_r}}$$

$$(27)\quad \sigma_m = \frac{\sigma_{y\cdot p}}{f}\,\frac{1}{1 + \frac{\sigma_{y\cdot p}}{\sigma_n}\,\frac{\sigma_r}{\sigma_m}}$$

The stress intensification factor operates to increase the nominal stress. The writer has chosen to consider that this factor operates to affect the strength of the pipe in the case of the variable stress only. In equations (25), (26) and (27) then σ_r should be replaced by $N\sigma_r$. Equations (26) and (27) then become

$$(28)\quad \sigma_r = \frac{\sigma_n}{N\,f}\,\frac{1}{1 + \frac{\sigma_n}{\sigma_{y\cdot p}}\,\frac{\sigma_m}{N\,\sigma_r}}$$

$$(29)\quad \sigma_m = \frac{\sigma_{y\cdot p}}{f}\,\frac{1}{1 + \frac{\sigma_{y\cdot p}}{\sigma_n}\,\frac{N\,\sigma_r}{\sigma_m}}$$

52. *The Method of Determining Safe Working Stresses Applied to Corrugated and Creased Pipes.*—In the great majority of prac-

tical applications a piping system will not be called upon to undergo the number of cycles of stress equivalent to that at which the endurance limit is determined. In the lifetime of an ordinary power plant, excluding serious vibration, the number of stress cycles imposed on the piping will probably be numbered in thousands rather than millions. It seems unreasonable then, in most cases, to use in design a stress value corresponding to the true endurance limit.

The true endurance limit represents the limiting value of stress to which a metal may be subjected in order to withstand an indefinitely large number of stress cycles. The fatigue graph defines this limit and also indicates the number of stress cycles to cause failure at stresses greater than the endurance limit. The value of stress taken from a fatigue graph for a stated number of cycles is here defined as the virtual endurance limit for the number of cycles selected. The definition for the stress intensification factor is extended to include the ratio of the virtual endurance limit of the steel to the virtual endurance limit of the pipe, the latter value again being expressed in terms of nominal stress. With the application limited to a stated number of cycles, the virtual endurance limit for the steel and the stress intensification factor at this virtual endurance limit may be substituted for σ_n and N in the formulas developed above.

The method used in determining the virtual endurance limits of the steel was similar to that described for determining the true endurance limits. At various selected cycle values, the stresses read from the fatigue graph of Plate VIII were expressed as certain percentages of the tensile strength of the steel used in determining that fatigue curve. These percentages were then applied to the experimentally determined tensile strengths of the pipe steel to establish the virtual endurance limits at the selected cycle values. This procedure was followed considering each series of piping individually. The stress intensification factors were then readily determined by dividing the virtual endurance limits of the steel by the virtual endurance limits of the pipe at selected cycle values. Equations (28), (29) and (24) were then solved to determine the value of the safe working stress for each series of pipes. The safe working stress was computed for two conditions; one assuming that the factor of safety was 2, and the other that the factor of safety was 1.5. The results of these computations are tabulated below:

Safe Working Stresses for Series A - Corrugated Pipe

Average tensile strength = 58400 lbs.per sq.in.
Average yield point = 38200 lbs.per sq.in.

No. of Cycles	Stress Intensification Factor N	Virtual end. Limit σ_n	Safe Working Stress for f = 2 σ max.	Safe Working Stress for f = 1.5 σ max.
$1x10^7$	8.7	25100	2680	3580
$5x10^5$	7.2	28800	3640	4840
$1x10^5$	5.2	34000	5580	7430
$5x10^4$	4.7	37500	6590	8800
$3x10^4$	4.5	40600	7300	9720
$2x10^4$	4.4	43500	7860	10460

Safe Working Stresses for Series B - Corrugated Pipe

Average Tensile Strength = 57300 lbs.per sq.in.
Average Yield Point = 39100 lbs.per sq.in.

No.of Cycles	Stress Intensification Factor N	Virtual Endurance Limit σ_n	Safe Working Stress for f = 2 σ max.	Safe Working Stress for f = 1.5 σ max.
$1x10^7$	9.5	24600	2430	3240
$5x10^5$	6.4	28300	3980	5300
$1x10^5$	4.3	33400	6480	8620
$5x10^4$	3.9	36800	7620	10160
$3x10^4$	3.8	40000	8280	11040
$2x10^4$	3.7	42600	8880	11840

Safe Working Stresses for Series C - Creased Pipe

Average Tensile Strength = 59000 lbs.per sq.in.
Average Yield Point = 40900 lbs.per sq.in.

No.of Cycles	Stress Intensification Factor N	Virtual Endurance Limit σ_n	Safe Working Stress for $f = 2$ $\sigma_{max.}$	Safe Working Stress for $f = 1.5$ $\sigma_{max.}$
1×10^7	6.1	25300	3770	5020
5×10^5	5.5	29000	4670	6220
1×10^5	4.5	34300	6430	8570
5×10^4	4.2	37800	7370	9820
3×10^4	3.9	41000	8340	11100
2×10^4	3.8	43800	9010	12000

These results indicate that, in general, the Series B corrugated pipe could be subjected to slightly higher safe working stresses than the Series A pipe. All pipes were manufactured with the same amplitude and pitch of corrugations. The Series B pipe, however, had a slightly greater wall thickness and a somewhat different curvature of corrugation from the Series A pipe. Typical sections of the two corrugated shapes are shown in Plate II and Figure 5 of Plate XXIX. It should be noted here that the flexibility factors for the tangent lengths of piping of each of the two series are essentially the same.

53. *The Selection of Safe Working Stresses for Design Purposes.*—The safe working stresses for each of the series of pipe may be expressed as a fractional part of the tensile strength as follows:

Ratio of Tensile Strength to Safe Working Stress

Cycles	Series A Corrugated		Series B Corrugated		Series C Creased	
	f = 2	f = 1.5	f = 2	f = 1.5	f = 2	f = 1.5
1×10^7	21.8	16.3	23.5	17.7	15.7	11.7
5×10^5	16.0	12.1	14.4	10.7	12.6	9.5
1×10^5	10.5	7.9	8.8	6.6	9.2	6.9
5×10^4	8.9	6.6	7.5	5.6	8.0	6.0
3×10^4	8.0	6.0	6.9	5.2	7.1	5.3
2×10^4	7.5	5.5	6.5	4.8	6.5	4.9

The average tensile strength for the steel used in the manufacture of the pipes tested was approximately 58,000 pounds per square inch, which is about the mean of the range of tensile strengths of steel used in ordinary commercial practice for seamless drawn steel tubing intended for high temperature service.* These ratios may then be used as a satisfactory approximation for design purposes in determining the safe working stress for pipes fabricated from tubing whose tensile strength and yield point values are within this usual commercial range.

The use of a factor of safety of 2 is considered conservative. The use of a lower value, such as 1.5 or some intermediate value, would undoubtedly be justified when it was known that the tubing material was of uniformly high quality, when the service conditions were not severe, and the values of the applied stresses could be predicted with reasonable accuracy.

Pipe shapes may successfully withstand large deflections and absorb very high applied stresses for single, or very few, applications of load. Mention has already been made of a test case where a pipe bend was completely collapsed without fracture. There may be some piping installations where the service application may involve an approximately steady loading condition. When this is

* See A. S. T. M. Specifications A-106-33T.

true the piping design may be based entirely on the static tensile properties of the metal. If, however, in the designer's opinion, the piping is to be subjected to variable stresses static strength considerations are insufficient and the design should be based on an extremely liberal allowance for the number of stress cycles to be encountered.

The fatigue graph is useful primarily for determining the safe limiting value of stress for a large number of cycles. The curve is necessarily the mean path through a number of observed points and can not be considered as applying with sufficiently close accuracy for literal application in predicting individual failures. A slight variation in stress value will cause a large percentage error in the number of cycles at any point on the curve. These considerations are particularly important in the high stress-low cycle region of the fatigue graph. The action of steel at high variable stresses and high temperatures can not be safely predicted. For these reasons the writer is of the opinion that 20,000 cycles of variable stress should be the absolute minimum basis for design purposes when cyclic stress conditions are considered, regardless of the actual predicted numerical value of the stress cycles to which the pipe may be expected to be subjected. In any case the number of cycles selected as a design basis, in accordance with the method presented, should be considerably greater than the number of cycles it is estimated the pipe will experience. This is particularly important, as has been previously indicated, in applications where corrosion is likely to be a factor.

54. *Application of Design Values to Other Sizes of Corrugated and Creased Pipes.*—Although many of the conclusions reached in this investigation are of general application, the values given for flexibility factors and for limiting design stresses will not be strictly applicable where piping is being considered which departs from geometrical similarity with the pipes tested. A flexibility factor value of between 5 and 6 will probably closely fit any usual corrugated pipe. The true value, however, can easily be determined by simple experiment.

Where piping sizes or creased or corrugated shapes are being considered whose dimensions are such as to make the factors for determination of safe working stresses here given of doubtful ap-

plication, the methods of stress determination based on strain measurement may be applied. For this purpose the dimensions of the pipe and corrugations or creases must be known.

Since the critical zone of failure in a creased pipe is on the outer wall surface the strain measurement method which has been described as applying to a corrugated pipe, may here be more simply applied. After readings with the bending strain gage and with the tensometer have been obtained, these values are used in connection with equations (1), (2), (16) and (17) to obtain results for the resultant longitudinal and hoop strains. The principal stresses and equivalent stress may then be determined in a manner similar to that described for the case of a corrugated pipe.

The yield point and tensile strength of the steel may be easily determined. A value of the virtual endurance limit for the steel is then selected, as has been described, for the desired number of cycles. Equations (24), (26) and (27) are then employed to determine the value for the safe working stress. The remaining step is to determine what loads, or deflections, will produce a value of the equivalent stress equal to this safe working stress.

XV. CONCLUSIONS.

55. *The Flexibility of Corrugated and Creased Bend Pipes.*—The load-deflection characteristics for each of the three series of corrugated and creased bend pipes are presented in Plates XIII to XVII. The flexibility factor for the corrugated tangents was found to be approximately 6. The corrugated arc compared to a theoretical smooth pipe arc of the same major dimensions had a relative flexibility advantage of 2.6 to 1. The corrugated arc compared to a smooth pipe arc is relatively less flexible than the corrugated tangent compared to a smooth pipe tangent because of the increased flexibility of a smooth pipe arc due to the flattening effect.

The creased bends had a relative flexibility compared to a theoretical smooth pipe bend of 1.3 to 1. Both the corrugated arc and the creased bend, however, were curved to a smaller radius of curvature than would be practicable for an actual smooth pipe bend.

The flexibility of both corrugated and creased pipes appeared to be unaffected by 400 pounds per square inch internal hydrostatic pressure.

56. *The Endurance Properties of the Pipes Under Variable Stresses.*—The results of the fatigue tests on each of the series of pipes are shown in Plates XIX to XXI, inclusive. Internal hydrostatic pressure of 400 pounds per square inch was applied to three pipes tested under cyclic stress and appeared to have no effect on the strength of the pipes.

The stress intensification factors at the endurance limits of the pipes were found to be—

Series A, $N = 8.7$
Series B, $N = 9.5$
Series C, $N = 6.1$

These results indicate that, at the endurance limit of the pipes, the nominal stresses which were computed for the given loading on a hypothetical smooth-walled pipe of the same basic dimensions as the pipes tested, were effectively multiplied by these factors. The values of the stress intensification factors for cycle values below the endurance limit are tabulated in section 52.

With one exception all fatigue failures in corrugated pipe developed from cracks which started from the inside surface of the pipe and worked outwards. The failures all occurred in the crest of the corrugations. The single exception concerned the failure of an expansion U-bend which failed from the outside between two corrugations, due probably to severe crimping of the pipe on the inner side of the arc where failure occurred, during fabrication.

Without exception all crease-bend failures occurred from cracks which developed from the outside pipe surface. These failures all occurred in the crease.

The nature of the failure occurring in the corrugated pipes indicates that it is unlikely that the ultimate fracture would be anticipated by the appearance of surface cracks on the outside of the pipe.

57. *The Measurement of Strains and the Determination of Stresses.*—A method is presented for measuring the bending strains and resultant strains on the outside of a corrugated pipe, and from these measurements to determine the strains on the inside surface of the pipe. The strains on the inside of these pipes are of greatest interest because of the indication from the fatigue tests that failure starts from this surface. It is further demonstrated how the prin-

cipal stresses may be determined from the strains. The maximum-shear theory is considered to apply to fatigue failures of pipes. The longitudinal stress, in general, appears to be the maximum principal stress on the inner surface of the corrugated pipe at the zone of failure of the radial stress the minimum principal stress. In most cases the hoop stress, being of intermediate value, will not affect the strength of the pipe. An investigation of the stresses caused by internal pressure indicated that, as confirmed by fatigue experiments, the internal pressure should have little or no effect on the ability of the pipes to withstand cyclic stresses.

58. *The Derivation of Safe Working Stresses.*—A method of determining safe working stresses based on the endurance limit and the yield point is presented. The method was applied to each of the series of pipes and results tabulated in section 52. Safe working stresses were determined using the true endurance limit and also using a virtual endurance limit as a basis. The use of this latter value yielded results for safe working stresses for various selected values of the number of stress cycles to which the pipe would be subjected.

For design purposes a tabulation is presented in section 53 from which, given the tensile strength of usual commercial seamless-drawn tubing, the safe working stresses for this tubing in the form of corrugated or creased pipes may be closely approximated.

It is considered that, if cyclic stress considerations are to enter into the design of pipe, 20,000 cycles of stress should be the absolute minimum used as a basis for design regardless of the actual predicted numerical value of the stress cycles which the pipe may be expected to experience. In any case the number of cycles selected as a design basis should be considerably greater than the number of cycles it is estimated the pipe will experience. This is highly important in applications where corrosion is likely to be a factor. A method is presented for determining safe working stresses for pipe shapes or sizes which deviate sufficiently from the pipes herein investigated to cause the factors presented to be of questionable suitability. This method involves the strain measurement methods presented.

59. *General Summary.*—The use of corrugated and creased pipe offers two principal advantages; the forces exerted against the

anchor points by pipe expansion are low, and the pipe may be bent to much smaller radii of curvature than is practicable for smooth-walled pipe. However, the factors which operate to increase the flexibility of these pipes in turn operate to increase the stress. This increase of stress may more than counterbalance the effect of increased flexibility.

It is essential that, since these pipe shapes involve areas of highly localized stress, the tubing from which the pipes are fabricated should be of uniformly high quality.

The primary objects of the investigation, stated in the Introduction, have been accomplished. The inherent complexity of the problem has made it impracticable to develop a general method for the determination of safe working stresses which will relieve the pipe designer from the necessity for exercising judgment. Methods have been developed, however, and factors presented which when properly applied should lead to a safe and satisfactory design of piping systems which include the corrugated and creased bend pipes.

APPENDIX.

I.

60. *Calculation of Bending Moments Due to Expansion in a Single Offset Quarter Bend.*—The calculations herein presented are for the purpose of illustrating the application of methods which have been adequately explained elsewhere.* For illustrative purposes the single offset quarter bend, piece AM, Plate III, has been selected. This bend is unsymmetrical and includes arcs of less than 90 degrees.

The purpose of the calculations is, for assumed conditions, to determine the bending moments at various parts of the bend. When the bending moments are determined, the value of the nominal stress at the point of maximum moment or at any other point may readily be calculated.

In this example the bend is assumed to be installed with the end flanges fixed. The bend is then considered to be heated. The anchor at one end of the bend is replaced by the forces necessary

* Piping Handbook by Walker and Crocker; discussion by A. E. R. de Jonge, of "Frictional Resistance and Flexibility of Seamless Tube Fittings Used in Pipe Welding," by Crocker and McCutchan, FSP-53-17, A. S. M. E. Trans., 1931. Also, Appendix IV to "Load-Deflection Relations for Large, Plain, Corrugated and Creased Pipe Bends," by Cope and Wert, FSP-54-12, A. S. M. E. Trans., 1932.

to maintain the flange in its fixed position. Figure 1 of Plate XLIII shows a diagrammatic dimensioned sketch illustrating this condition.

The following assumptions are made:

Temperature rise of pipe = 400 degrees F., from 32 degrees F.

$E = 30 \times 10^6$ pounds per square inch at 32 degrees F.

Outside diameter of basic pipe = 6.125 inches.

Inside diameter of basic pipe = 5.779 inches.

Dimensions of the bend are as shown in Figure 1, Plate XLIII.

Modulus of Elasticity at 432 degrees F.

$$E_t = E_{32}\left[1 - \left(\frac{t - 32}{1700}\right)^2\right]*$$

(30) $E_t = 30 \times 10^6 \times 0.945 = 28.35 \times 10^6$ pounds per square inch.

Linear Expansion at 432 degrees F.

From the formula of Holborn and Day (Mark's Handbook, 2nd Ed., pg. 300):

$$l = l_o\left[1 + a\left(\frac{t - 32}{1000}\right) + b\left(\frac{t - 32}{1000}\right)^2\right]$$

where l = length of metal at temperature t.

l_o = length of metal at 32 degrees F.

$a = .006212$ for steel

$b = .001623$ for steel

Hence the inches increase in length per inch of metal at 400 degrees F. would be:

(31) $l = 2746 \times 10^{-6}$ inches

The Rigidity Multiplication Factor for a Smooth Pipe of Equivalent Basic Dimensions.

$$K = \frac{10 + 12\lambda^2}{1 + 12\lambda^2}$$

$$\lambda = \frac{tR}{r^2}$$

---

* "High Pressure Steam Boilers," by G. A. Orrok, A. S. M. E. Trans., 1928.

where t = wall thickness in inches = 0.173
R = radius of arc in inches = 15
r = mean radius of pipe in inches = 2.976
$\lambda = 0.293$

and

(32) K = 5.433

The Moment of Inertia and the Section Modulus.

$$I = \frac{\pi}{64} (D^4 - d^4)$$

(33) I = 14.33

The section modulus, I/c, is equal to the above value of the moment of inertia divided by the mean pipe radius 3.063 and has the value of 4.68.

Figure 2 of Plate XLIII represents the bend broken up into its elements, with the force Fy acting. The moment diagram resulting from the application of this force is shown in Figure 3. The area enclosed in the dotted lines represents the resisting moment due to the fixed end assumption. The ordinate, 31.4 F, is equal to the net area of the bending moment diagram due to Fy, divided by the length of the diagram. The determination of the value of this ordinate presupposes a knowledge of the value of V, the flexibility factor. This factor has been developed for standard shapes as explained in sections 27 to 31. The corrugation shape in this bend differed somewhat from the others tested and the flexibility factor, as will be shown, was determined by actual experiment on this bend. The case numbers on Figure 3 refer to Plates XXXIX and XL. These plates contain diagrams taken from sources previously mentioned.* Plate XLI contains curves of deflection factors to be used in connection with Plate XL.

Calculation for $\delta y = f(F_y)$ [Figure 3, Plate XLIII].

$EI\delta_{ye} = A_b \times 1 = 413\ F \times 3.3$

(From rotation) 1 363 F Up

$EI\delta_{ym}$ = Case XIII = 344.3 VF × .145 × 15 (from moment) 749 VF Up

* Piping Handbook and A. S. M. E. Transactions FSP-54-12 loc. cit.

$EI\delta_{yf}$ = Case XI = 22.5 VF × .180 × 15			
(From force)			61 VF Down
$EI\delta_{yd} = A_c \times 1 - (413\ F + 321.8\ VF)\ 3.3$	1	363	F Up
	1	005	VF Up
$EI\delta_{ym}$ = Case XVI = 378 VF × .077 × 15		291	VF Up
$EI\delta_{yf}$ = Case XIV = 11.6 VF × .135 × 15		24	VF Down
$EI\delta_{ye} = A_d \times 1 = (413\ F + 688.2\ VF)\ 15$	6	195	F Down
	10	323	VF Down
$EI\delta_{ym}$ = 518.5 VF × .637 × 15	5	000	VF Down
$EI\delta_{yf}$ = Case V = 225 VF × .785 × 15	2	655	VF Down
$EI\delta_{yf} = A \times 1 - (413\ F + 1432\ VF)\ 18$	7	434	F Down
	25	776	VF Down
$EI\delta_{ym}$ = 72 F × 9		648	F Down
$EI\delta_{yf}$ = 162 F × 12	1	944	F Down

The summation of $EI\delta_y$ gives, in the down direction—

(34) $\Sigma EI\delta_y = 13{,}495 + 41{,}794\ VF$

Plate XLII shows the load-deflection relationship for this bend to be $\delta_y = 0.9 \times 10^{-3}$ F for an applied downward force. This measurement was made at room temperature for which $E = 30 \times 10^6$. Substituting this value of E and the experimental value of δ_y in 34, a value for V is obtained which is independent of pipe temperatures.

(35) $V = 8.935$.

It is interesting to note here from the previously computed value for K (eq. 32) that $V/K = 1.64$ or this bend is 64 per cent more flexible than a theoretical smooth pipe of the same dimensions bent to the same arc radius.

then

(36) $\delta_y = \frac{952.0}{10^6} F_y$ (From 35, 30 and 33 in 34)

Calculation for $\delta_x = f(F_y)$ [Figure 3, Plate XLIII].

$EI\delta_{xb} = A \times 1/2 = 413\ F \times 6.8$	2	808	F Right
$EI\delta_{xc} = A_b \times 9.44 = 413\ F \times 9.44$	3	899	F Right
$EI\delta_{xf}$ = Case XI = 22.5 VF × .375 × 15		127	VF Left

$EI\delta_{xm}$ = Case XIII = 344.3 × .301 × 15	1	554 VF	Right
$EI\delta_{xd} = A_c \times 1$ = (413 F + 321.8 VF) 9.44	3	899 F	Right
	3	038 VF	Right
$EI\delta_{xf}$ = Case XIV = 11.63 VF × .490 × 15		086 VF	Left
$EI\delta_{xm}$ = Case XVI = 378 VF × .330 × 15	1	871 VF	Right
$EI\delta_{xe} = A_d \times 1$ = (413 F + 688.2 VF) 15	6	195 F	Right
	10	323 VF	Right
$EI\delta_{xf}$ = 225 VF × .5 × 15	1	688 VF	Right
$EI\delta_{xm}$ = 518.5 VF × .363 × 15	2	833 VF	Right

The summation of $EI\delta_x$ gives to the right

(37) $\Sigma EI\delta_x = 16{,}801\ F + 21{,}094\ VF$

Hence

(38) $\delta_x = \frac{505.3}{10^6} F_y$ (From 35, 30 and 33 in 37).

Figure 3 of Plate XLIII shows this bend broken up into its elements and acted upon by the force F_x. Figure 5 is the moment diagram resulting from this force. The area enclosed by dotted lines is again the value of restraining moment.

Calculations for $\delta_x = f(F_x)$ [Figure 5, Plate XLIII].

$EI\delta_{xf} = A_f \times 21/3 = 92.48\ F \times 9.07$		839 F	Left
$EI\delta_{xm} = A_m \times 1/2 = 460.8 \times 13.6\ F$	3	133 F	
$EI\delta_{xc} = A \times 1 = 553 \times 9.44$	5	220 F	
$EI\delta_{xf}$ = Case XII = $A_f \times .415\ R$ = 46.31 VF × .415 × 15		288 VF	
$EI\delta_{xm}$ = Case XIII = $A_m \times .301 \times R$ = 249.6 VF × .301 × R	1	127 VF	
$EI\delta_{xd} = A_c \times 1$ = (295.9 VF + 553.3 F) 9.44	2	793 VF	
	5	223 F	
$EI\delta_{xf}$ = Case XV = $A_f \times .433 \times R$ = 50.18 VF × .433 × 15		326 VF	
$EI\delta_{xm}$ = Case XVI = $A_m \times .330 \times R$ = 153.2 VF × .33 × 15		758 VF	

$EI\delta_{xe} = A_d \times 1 = (499.3\ VF + 553.3\ F)\ 15$ 7 490 VF
8 300 F

$EI\delta_{xf} = A \times .624 \times R = 128.25\ VF \times .624 \times 15$ 1 200 VF

A summation of $EI\delta_x$ gives to the left

(39) $\Sigma EI\delta_x = 22{,}715\ F + 13{,}982\ VF$

then

(40) $\delta_x = \frac{363.4}{10^6} F_x$ (From 30, 33 and 35 in 39)

Calculations for $\delta_y = f\ (Fx)$

$EI\delta_{ye} = A_b \times 1 = 553.3\ F \times 3.3$ 1 826 F Down

$EI\delta_{yf} =$ Case XII $= A \times .195 \times R = 46.31\ VF \times .195 \times 15$ 136 VF Down

$EI\delta_{ym} =$ Case XIII $= A \times .145 \times R = 249.6 \times .145 \times 15$ 543 VF Down

$EI\delta_{yd} = A_c \times I = (295.9\ VF + 553.3\ F) \times 3.3$ 977 VF Down
1 826 F Down

$EI\delta_{yf} =$ Case XV $= 50.18\ VF \times .115 \times 15$ 087 VF Down

$EI\delta_{ym} =$ Case XVI $= 153.2\ VF \times .077 \times 15$ 177 VF Down

$EI\delta_{ye} = A_d \times 1 = (499.3\ VF + 553.3\ F)\ 15$
7 490 VF Up
8 300 F Up

$EI\delta_{yf} = 128.3\ VF \times .876 \times 15$ 1 686 VF Up

$EI\delta_{yg} = A_e \times 1 = (627.6\ VF + 553.3\ F)\ 22$
13 807 VF Up
12 173 F Up

A summation of $EI\delta_y$ gives, in the up direction

(41) $\Sigma EI\delta_y = 16{,}821\ F + 21{,}063\ VF$

then

(42) $\delta_y = \frac{504.6}{10^6} F_x$ (From 30, 33 and 35 in 41)

Adopting the convention that directions upwards and to the right are positive

$$(43)\quad \delta_x = \frac{363.4}{10^6} F_x - \frac{505.3}{10^6} F_y \text{ (From 38 and 40)}$$

$$(44)\quad \delta_y = -\frac{504.6}{10^6} F_x + \frac{952}{10^6} F_y \text{ (From 36 and 42)}$$

If the bend is assumed to be heated to 432 degrees F., the deflections to be forced back into the bend are the vertical length of the bend in the direction times the coefficient of expansion of steel at 432 degrees F. and the horizontal length of the bend in the x direction times the same coefficient.

$$(45)\quad \delta_x = 34.25 \times 2746 \times 10^{-6} = 94{,}050 \times 10^{-6}$$
(From 31)

$$(46)\quad \delta_y = 47.5 \times 2746 \times 10^{-6} = 130{,}400 \times 10^{-6}$$
(From 31)

Then

$$(47)\quad 94{,}050 = 363.4\, F_x - 505.3\, F_y \text{ (From 45 in 43)}$$

$$(48)\quad 130{,}400 = -504.6\, F_x + 952.0\, F_y \text{ (From 46 in 44)}$$

Solving these equations as a simultaneous system—

$$(49)\quad F_x = 1716$$

$$(50)\quad F_y = 1046$$

These are the forces exerted by the pipe against the anchor. Hence the reactive forces which are exerted on the pipe are:—

$$(51)\quad F_x = -1716$$

$$(52)\quad F_y = -1046$$

The ordinates of the bending moment diagrams, Figures 3 and 5, Plate XLIII, represent the values of the moments in terms of the applied forces. Since these Figures are not drawn to scale and are illustrative only, the curves joining the ends of the ordinates are not exact. The values of the bending moments are defined from these Figures only at the points of beginning and ending of the elements of the bend.

Moments Due to F_y—From Figure 3, Plate XLIII

Position	F_y	*Net Ordinate*	*Bending Moment*
a	—1046	—1	+1046
b		—1	+1046
c		+2.3	—2406
d		+5.6	—5858
e		—9.4	+9832
f		—27.4	+28,660
g		—31.4	+32,844

Moments Due to F_x—From Figure 5, Plate XLIII

Position	F_x	*Net Ordinate*	*Bending Moment*
a	—1716	—33.13	+56,851
b		—19.53	+33,513
c		—10.13	+17,383
d		—0.53	+909
e		+14.37	—24,659
f		+14.37	—24,659
g		+14.37	—24,659

These Bending Moments due to F_x and F_y are then combined

Position	*Total Moment*
a	+57,900
b	+34,600
c	+16,000
d	—4750
e	—14,800
f	+4000
g	+8180

The values of these moments have been plotted against the value of the developed length of the bend as shown in Plate XLIV. From this diagram it will be seen that the maximum moment at the bend is at point a and that the maximum moment in a corrugated section is at point b. The minimum moment lies between points d and e.

The nominal stresses may be computed from the bending formula

$$S = \frac{Mc}{I}$$

or at point a

$$S = \frac{57{,}900}{4.68} = 12{,}400 \text{ pounds per square inch}$$

and at point b

$$S = \frac{34{,}600}{4.68} = 7390 \text{ pounds per square inch.}$$

The nominal stresses at any other point on the bend may be computed in a similar manner using the proper value for the bending moment, selected from the curve of Plate XLIV.

ACKNOWLEDGMENTS.

The work upon which this dissertation is based was undertaken at the U. S. Naval Engineering Experimental Station, Annapolis, Maryland, under the authorization and direction of the Bureau of Engineering of the Navy Department. The writer wishes to acknowledge his indebtedness to Captain O. L. Cox, U. S. Navy, Director of the Experiment Station, for his active interest and constant encouragement in carrying out the investigation.

The writer also wishes to take this opportunity to express his thanks to others who have aided him. Mr. W. C. Stewart, Metallurgist, has contributed many valuable suggestions and has given much helpful advice. Dr. W. M. Coates, Associate Professor of Mathematics and Mechanics at the U. S. Naval Postgraduate School, has contributed valuable criticism. He suggested the method of measuring bending strains in the pipe and has kindly permitted the writer to refer to his unpublished monograph on stress distribution in shells. Mr. B. F. Treat, Associate Mechanical Engineer, has contributed helpful suggestions and assisted in designing the fatigue testing machines and in checking the computations. Mr. J. D. Irons, Junior Metallurgist, and Mr. R. W. Reynolds, Under Engineering Aide, performed excellent work in observing many of the

experimental data. Mr. J. B. Lincoln, Senior Mechanical Engineer, has also contributed valuable suggestions.

Appreciation and gratitude are due Professor A. G. Christie, Department of Mechanical Engineering, of the Johns Hopkins University, for his inspiring instruction in engineering, and for his helpful interest and criticism in connection with this investigation.

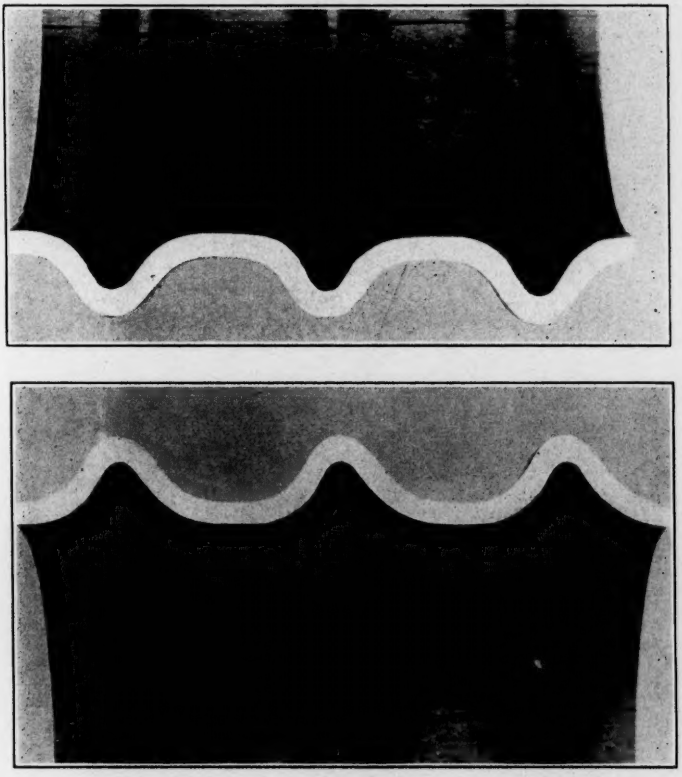


PLATE I.

FIG. 1 (AT BOTTOM)—LONGITUDINAL SECTION SHOWING A PROFILE OF TYPICAL CORRUGATIONS IN SERIES A PIPING.

Note the slightly different curvatures of the corrugations at the inner pipe surface.

FIG. 2 (AT TOP)—LONGITUDINAL SECTION SHOWING A PROFILE OF TYPICAL CORRUGATIONS IN SERIES B PIPING.

Note here also the lack of exact similarity in the corrugation.

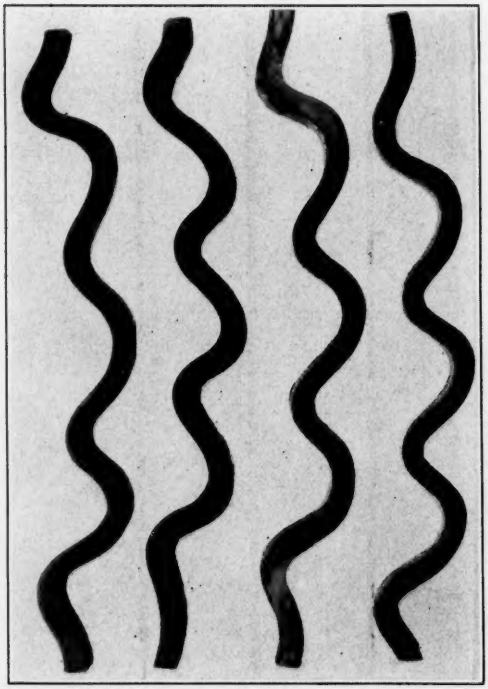


PLATE II—LONGITUDINAL STRIPS CUT FROM THE PLANE OF BENDING IN A CREASED-BEND.

These strips were cut from piece CD, an expansion U-bend. Each strip represents half an arc length of each of the four creased arcs. In this photograph the inner pipe surface is at the right of each strip. Note the individual character of the creases.

Type Bend	Designation	Sketch	Dimensions R	A	B	C	D	Class Steel A.S.T.M.	Physical Properties Tensile Strength	Yield Point	Elongation % 1 in.	Elongation % 8 in.	Red. Area %	Chemical Analysis C	S	P	Mn	Si	Heat Treatment (See Note)
Straight Tangent	AA			6'-9"	5'-9"			A	51,500	35,580	62	49	67						Normalized
	AB			6'-6"	5'-6"			A	51,650	34,350	62	48	68.4	0.13	0.024	0.011	0.53	0.01	Stress Relieved
	AC			6'-9"	5'-9"			B	68,200	43,900	51	36	43.4	0.21	0.029	0.044	0.36	0.012	Normalized
Expansion U-Bend	AD		1'-9"	8'-6"	6'-11"	6'-6"		A	58,000	35,350	64	45.5	65.1						Normalized
	AE		1'-9"	8'-6"	6'-10"	6'-6"		B	70,000	43,000		36	46						Normalized
	AF		1'-9"	8'-6"	6'-10"	5'-11"		A	54,700	38,100	64	45.5	68.9	0.16	0.022	0.013	0.54	0.10	Stress Relieved
Quarter Bend with Tangents	AG		1'-9"	5'-3"	5'-3"			A	52,300	33,400	51	33	62.9	0.13	0.024	0.011	0.53	0.01	Normalized
	AH		1'-9"	5'-3"	5'-3"			A	51,600	37,200	60	40	72.6						Stress Relieved
Straight Tangent (Plain Ends)	AJ			13'-0"	3'-4"			A	60,000	36,900		41.7							Stress Relieved
	AK			13'-6"	3'-4"			A	61,250	40,000		39							Stress Relieved
	AL			13'-10"	3'-0"			A	61,500	42,000	60	41.5	61						Stress Relieved
Single Offset Quarter Bend	AM		1'-6"	3'-11"	5'-6"	1'-6"	1'-6"												
			Special Semi-Corrugated Bend 6.125 in. OD × 0.173 in. Wall Thickness																

All Pipe - Seamless Steel Tubing
6.625 in. O.D. × .280 in. Wall Thickness
Pitch of Corrugations - 2½ in.
Amplitude of Corrugations - ¾ in.

Note - Heat Treatment
Normalized - 1575°F for 30 min. Cooled in Air
Stress Relieved - 1250°F for 30 min. Cooled in Furnace

PLATE III—MATERIAL LIST FOR SERIES A PIPING.

Type Bend	Designation	Sketch	Dimensions R	A	B	C	D	Class Steel A-106-33T	Physical Properties Tensile Strength	Yield Point	Elongation % 1 in.	Elongation % 2 in.	Red. Area %	Chemical Analysis C	S	P	Mn	Si	Heat Treatment (See Note)
Straight Tangent	BA			6'-3"	4'-9"			A	53,600	37,200	48.0	35.0	33.5						Stress Relieved
	BB			6'-3"	4'-9"			A	53,800	37,200	48.0	35.0	33.5	0.14	0.029	0.012	0.39	0.01	Stress Relieved
	BC			6'-4"	4'-9"			A	53,800	37,200	48.0	35.0	33.5						Stress Relieved
Quarter Bend with Tangents	BD			1'-3"	5'-3"	5'-3"		A	53,300	37,400	35.0	42.0	54.8						Stress Relieved
	BE			1'-3"	5'-3"	5'-2½"		A	53,750	38,300	38.0	36.5	67.4						Stress Relieved
	BF			1'-3"	5'-3"	5'-3"		A	53,750	38,300	38.0	36.5	67.4						Stress Relieved
Straight Tangent (Plain Ends)	BG			13'-10"	3'-4"			A	61,100	42,100	41.0	41.0	63.4						Tempered
	BH			13'-10"	3'-4"			A	72,600	46,000		25.5	75.1	0.17	0.024	0.015	0.66	0.15	Tempered
	BJ			13'-10"	3'-4"			A	57,500	38,300	54.0	57.5	54.8						Stress Relieved
Quarter Bend with Tangents	CA			1'-3"	5'-3"	5'-3"		A	54,900	41,900		41.0	66.1	0.16	0.022	0.015	0.66	0.15	Stress Relieved
	CB			1'-3"	5'-3"	5'-3"		A	53,800	35,250		40.2	60.0						Stress Relieved
	CC			1'-3"	5'-3"	5'-3"		A	56,500	40,000	40.0	39.0	61.7						Stress Relieved
Expansion U-Bend	CD			1'-3"	8'-3"	3'-11"	4'-0"	0'-11"	A	63,000	42,100		32.8	68.6						Stress Relieved
	CE			1'-3"	8'-4"	4'-0"	4'-0"	1'-9"	A	60,000	43,700	39.0	42.0	66.9						Stress Relieved
	CF			1'-3"	8'-4"	4'-0"	4'-0"	1'-9"	A	60,500	42,300	46.0	38.5	68.6						Stress Relieved

All Pipe - Seamless Steel Tubing
6.625 in. O.D. × 0.288 in. Wall Thickness
Pitch of Corrugations - 2½ in.
Amplitude of Corrugations - 1 in.
Pitch of "Creases" - 11° 15' approx.

Note - Heat Treatment
Stress Relieved - Heated to 1200 °F for 10 min. Cooled in Air.
Tempered - Heated to 1800 °F. Quenched in Water
Reheated to 1200 °F. Cooled in Air.

PLATE IV—MATERIAL LIST FOR SERIES B PIPING.

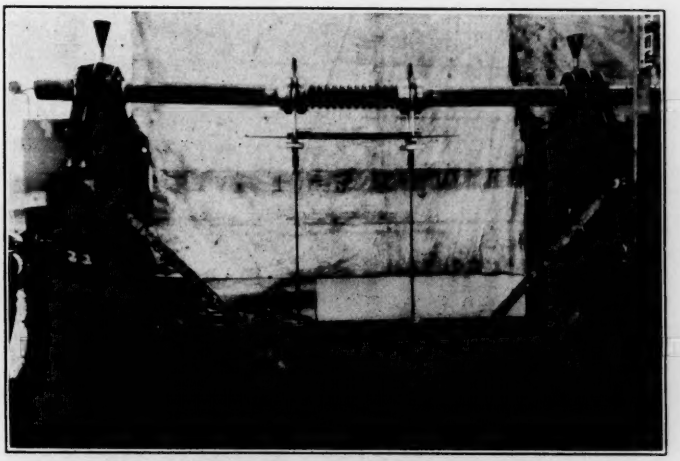


PLATE V—THE ROTATING BEAM TESTING MACHINE.

This machine was designed to apply a constant loading, or specific stress, to the pipe specimen. The pipe was rotated about its axis by a belt drive. A desired amount of dead weight load could be applied on the hangers shown below each journal at the ends of the corrugated section. Note the six-foot scale extended below the corrugated pipe section.

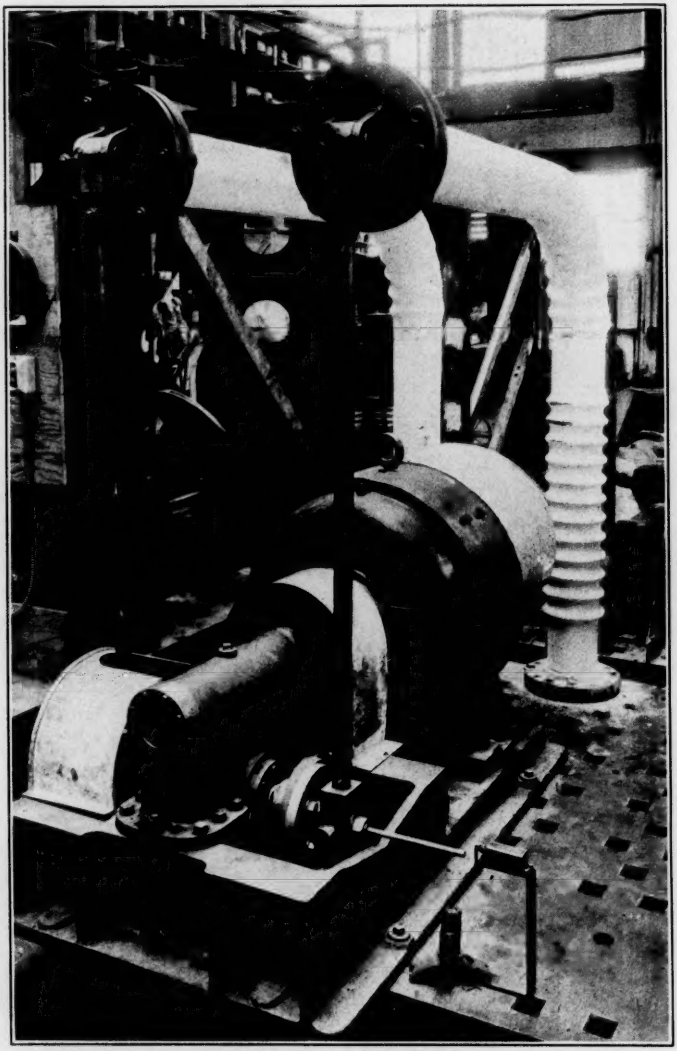


PLATE VI—TWO QUARTER-BENDS ERECTED FOR TEST ON THE MOTOR DRIVEN SPECIFIC STRAIN TESTING MACHINE.

The motor drove, through reduction gearing, two cranks mounted on eccentrics. The eccentrics were adjustable to produce the desired amount of deflection of the pipes.

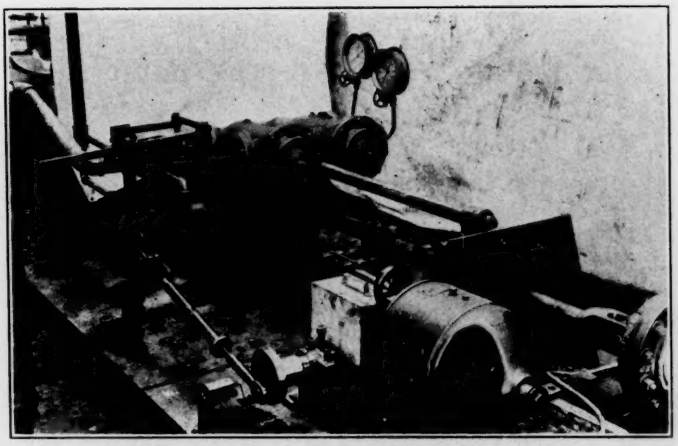


PLATE VII—PUMP DRIVEN SPECIFIC STRAIN TYPE TESTING MACHINE.

This machine was developed from the water end of a reciprocating pump. A hydraulic system, under pressure, was connected to either end of the pump cylinder. The water was alternately admitted to one end of the cylinder and discharged from the other through an arrangement of plug cocks. The plug cocks were operated by a motor driven through linkages, shown in foreground. A flange of a pipe specimen under test may be seen in the lower right of the photograph, attached to the end of the pump rod. The movement of the rod was controlled by suitable adjustment of the plug cock linkage and by stops applied to the rod. The cycle frequency was about 20 per minute.

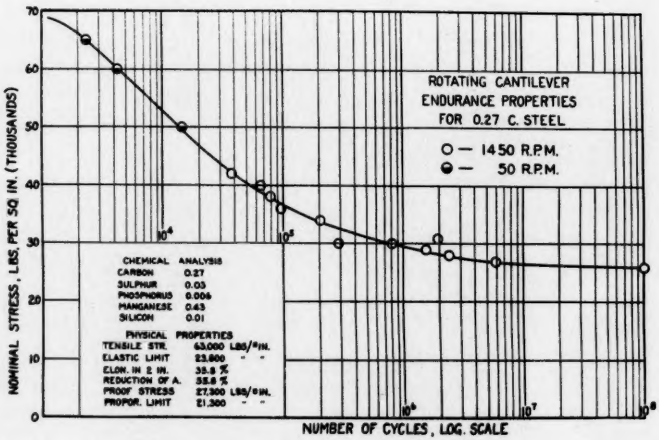


PLATE VIII—FATIGUE GRAPH FOR A LOW CARBON STEEL SIMILAR TO THE STEEL USED IN THE MANUFACTURE OF THE PIPE UNDER TEST.

The ordinate scale represents the value of the applied bending stress and the abscissa scale represents the number of cycles. The points on the graph indicate failures of specimens. Rotating cantilever specimens were used. These specimens had a conically tapered test length and a maximum diameter in the region of highest stress of about one-half inch.

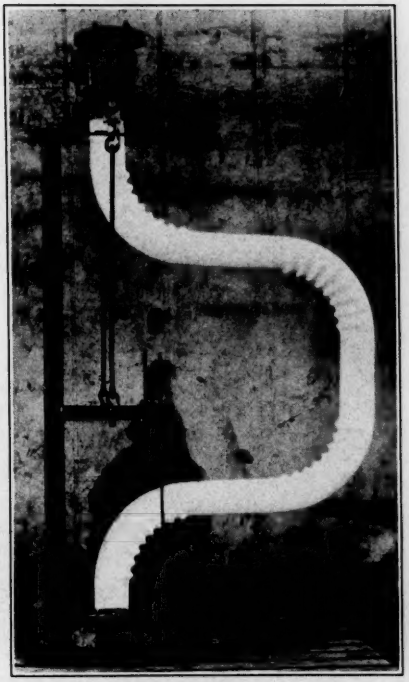


PLATE IX—AN EXPANSION U-BEND ERECTED FOR LOAD-DEFLECTION MEASUREMENTS IN THE FREE END CONDITION.

The deflections were measured from the top flange to the fixed framework mounted just below it. The loads were hung on pivoted supports from a cross-piece.

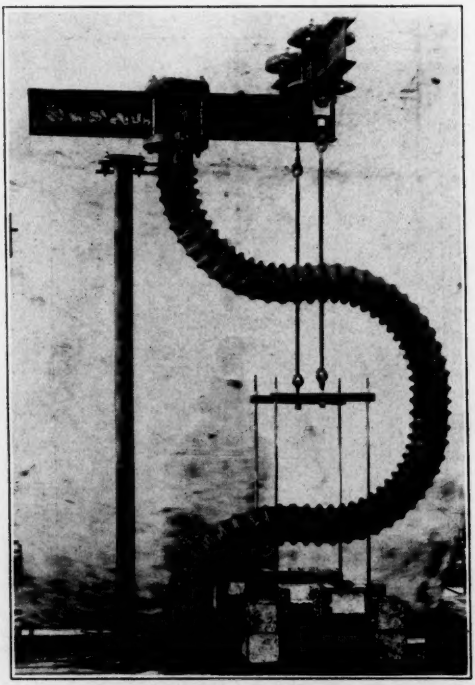


PLATE X—AN EXPANSION U-BEND ERECTED FOR LOAD-DEFLECTION MEASUREMENTS IN THE FIXED END CONDITION.

The moment arm from the center of the top flange to the line of action of the applied dead-weight load was adjusted so that the top flange always remained horizontal as increasing loads were applied. This condition was checked by a spirit level (not shown) mounted on the top flange. The deflections were measured from the top flange to a fixed framework just under the flange.

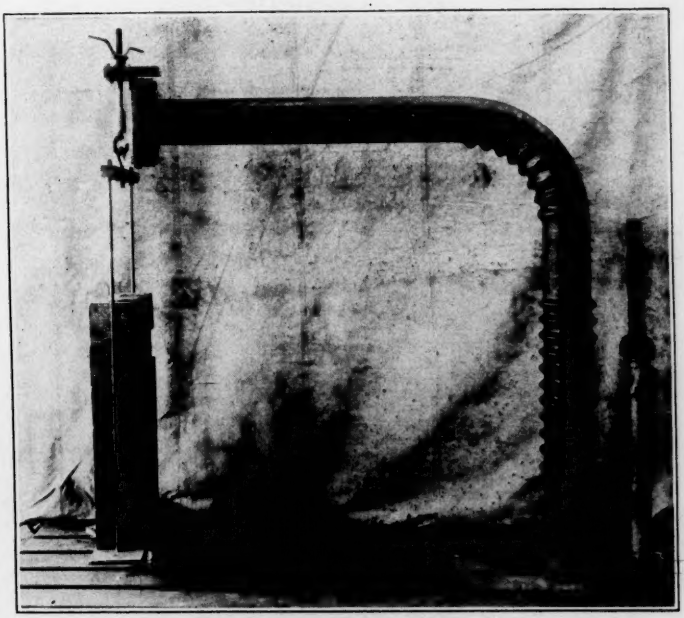


PLATE XI—A QUARTER-BEND ERECTED FOR LOAD-DEFLECTION MEASUREMENTS.

The deflections were measured from the bottom of the free flange to a fixed batten mounted just below it. Standard weights were supported in a pan hung from the free flange.

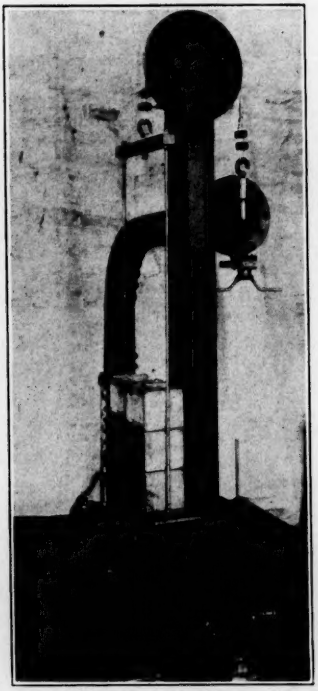


PLATE XII—A QUARTER-BEND ERECTED FOR LOAD-DEFLECTION MEASUREMENTS WITH THE LOADING APPLIED UPWARDS.

The deflections were measured from the free flange to a fixed batten. Standard weights were supported in a pan to apply the loading.

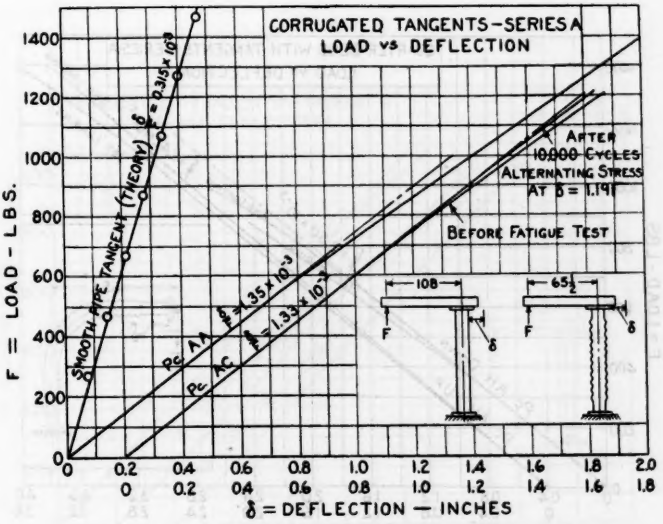


PLATE XIII.

FIG. 1—LOAD-DEFLECTION CURVES FOR CORRUGATED TANGENTS, SERIES A.

This figure shows the load-deflection relationship for pieces AA and AC. The diagram in the lower right corner indicates the method of loading and the location of the deflection measurements. On this figure there is also shown the results of load-deflection measurements on a smooth pipe of the same basic dimensions as the corrugated tangent. The line at the left of the figure represents the load-deflection relationship according to theory as a result of computations using the moment-area method. The circles along this line are experimentally determined points. This demonstrates the exceedingly close agreement between the theoretical and experimental results.

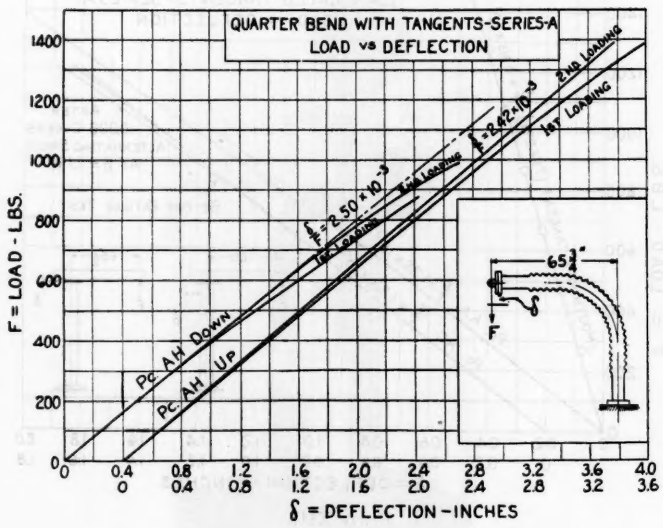


PLATE XIII.

FIG. 2—LOAD-DEFLECTION CURVES FOR CORRUGATED QUARTER-BENDS, SERIES A.

This figure shows the load-deflection relationship for piece AH. The diagram indicates the method of loading and the point of deflection measurements. The curves given represent the load-deflection for the pipe as received and again after a few stress cycles. It will be noted that the effect of the stress cycles is to raise the elastic limit of the pipe as a whole.

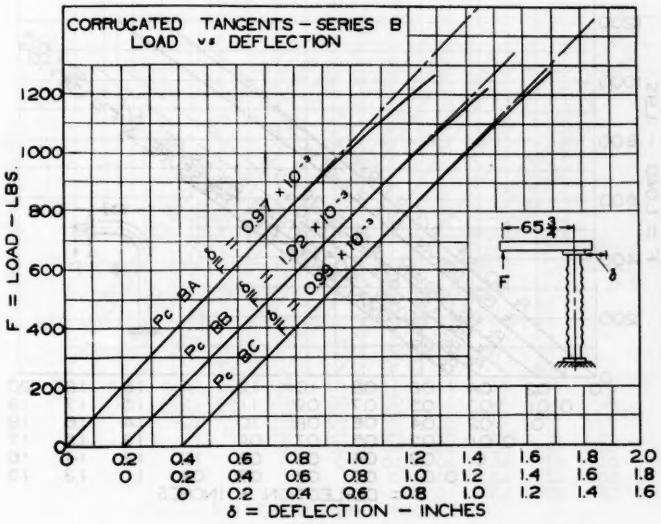


PLATE XIV.

FIG. 1—LOAD-DEFLECTION CURVES FOR CORRUGATED TANGENTS, SERIES B.

These curves show the load-deflection relationship for pieces B A, B B, and B C.

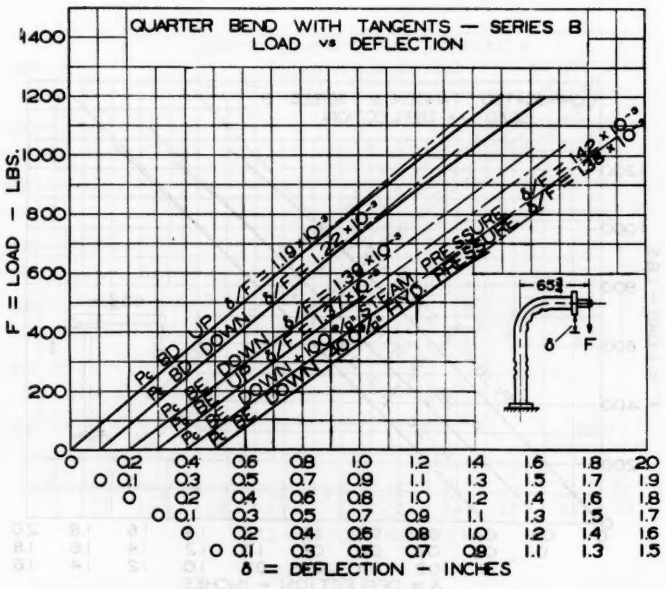

QUARTER BEND WITH TANGENTS — SERIES B
LOAD vs DEFLECTION
F = LOAD - LBS.
δ = DEFLECTION - INCHES
Pi BD UP δ/F = .119 x 10⁻³
Pi BD DOWN δ/F = 1.22 x 10⁻³
Pi BE UP δ/F = 1.39 x 10⁻³
Pi BE DOWN δ/F = 1.42 x 10⁻³
Pe BE UP δ/F = 1.36 x 10⁻³
Pe BE DOWN δ/F = 1.42 x 10⁻³
100 lbs/sq in STEAM PRESSURE
400 lbs/sq in HYD. PRESSURE
65 3/4
δ
F

PLATE XIV.

FIG. 2—LOAD-DEFLECTION CURVES FOR QUARTER-BENDS WITH CREASED ARC AND CORRUGATED TANGENT, SERIES B.

These curves show the load-deflection relationship for pieces B D and B E. Note that the load-deflection curve for piece B E under 400 pounds per square inch internal hydrostatic pressure is essentially the same as without internal pressure. The terms up and down applied to the curves, indicate the direction of the applied load at the free flange.

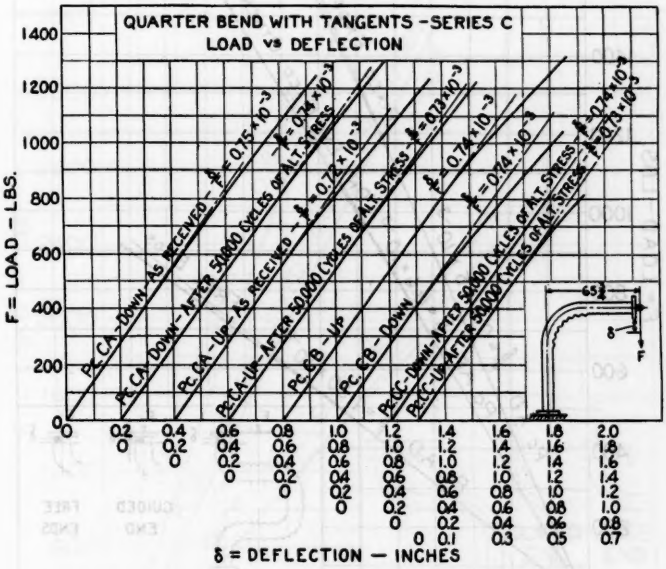

QUARTER BEND WITH TANGENTS - SERIES C
LOAD vs DEFLECTION
F = LOAD - LBS.
δ = DEFLECTION — INCHES
Pc. CA - DOWN - AS RECEIVED - f = 0.75 × 10⁻³
Pc. CA - DOWN - AFTER 50000 CYCLES OF ALT. STRESS - f = 0.74 × 10⁻³
Pc. CA - UP - AS RECEIVED - f = 0.72 × 10⁻³
Pc. CA - UP - AFTER 50,000 CYCLES OF ALT. STRESS - f = 0.73 × 10⁻³
Pc. CB - UP
Pc. CB - DOWN
Pc. CC - DOWN - AFTER 50000 CYCLES OF ALT. STRESS - f = 0.74 × 10⁻³
Pc. CC - UP - AFTER 50000 CYCLES OF ALT. STRESS - f = 0.73 × 10⁻³
65°
δ
F

PLATE XV—LOAD-DEFLECTION CURVES FOR QUARTER-BENDS WITH CREASED ARC, SERIES C.

These curves show the load-deflection relationships for pieces C A, C B and C C. The effect of cold work on the raising of the primitive yield point is shown by the raising of the proportional limit of these curves after the pieces had been subjected to cycles of alternating stress.

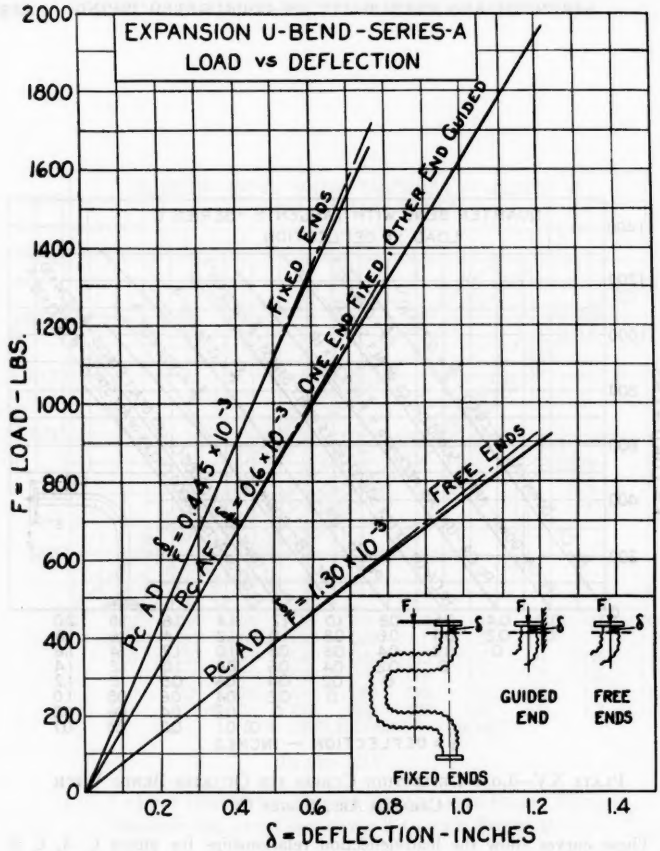


PLATE XVI—LOAD-DEFLECTION CURVES FOR EXPANSION U-BENDS, SERIES A.

These curves show the load-deflection relationship for pieces A D and A F. Three loading conditions are represented; fixed ends, one end fixed and the other end guided, and free ends. The free end and fixed end conditions are shown in Plates IX and X. For the guided end condition the U-bend was erected with the bend lying in a vertical plane in the pump type specific strain machine (Plate VII). The loading was applied, and measured, through a calibrated ring. In this test one end of the bend was fixed. The other end was attached to the pump rod through a horizontal pivot joint. The moving flange was thus allowed to rotate and its center was constrained by the pump rod to move in a horizontal line.

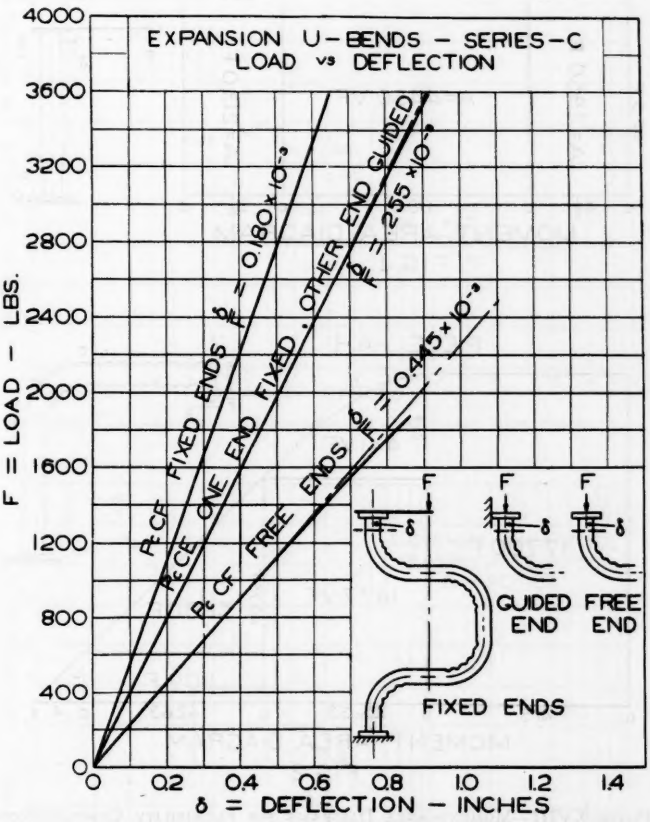


PLATE XVII—LOAD-DEFLECTION CURVES FOR EXPANSION U-BENDS, SERIES C.

These curves show the load-deflection relationships for Pieces C E and C F. The significance of the various loading conditions is explained in section 25.

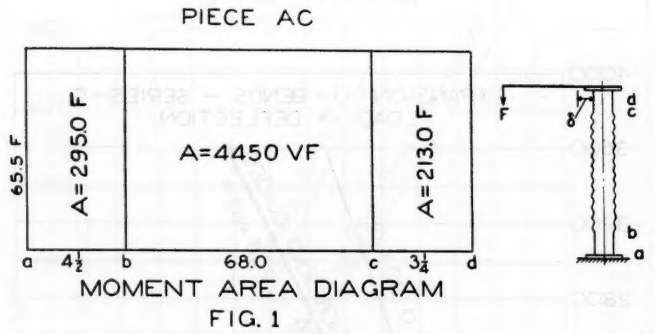


FIG. 1

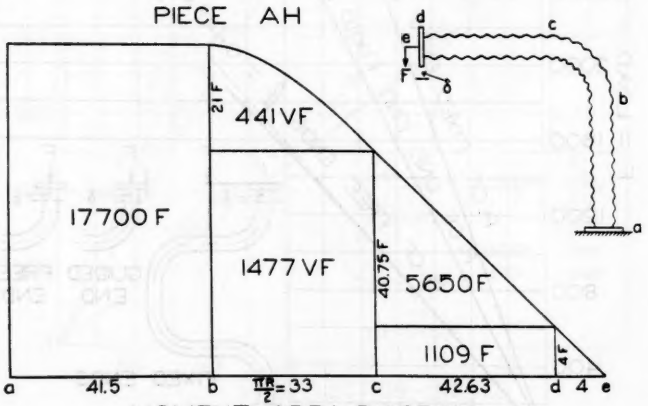


FIG. 2

PLATE XVIII—MOMENT-AREA DIAGRAMS FOR FLEXIBILITY COMPUTATIONS.

FIG. 1—MOMENT-AREA DIAGRAM FOR PIECE A C. THE COMPUTATION OF THE FLEXIBILITY FACTOR IS GIVEN IN SECTION 27.

FIG. 2—MOMENT-AREA DIAGRAM FOR PIECE A H. THE COMPUTATION OF THE FLEXIBILITY FACTOR IS GIVEN IN SECTION 28.

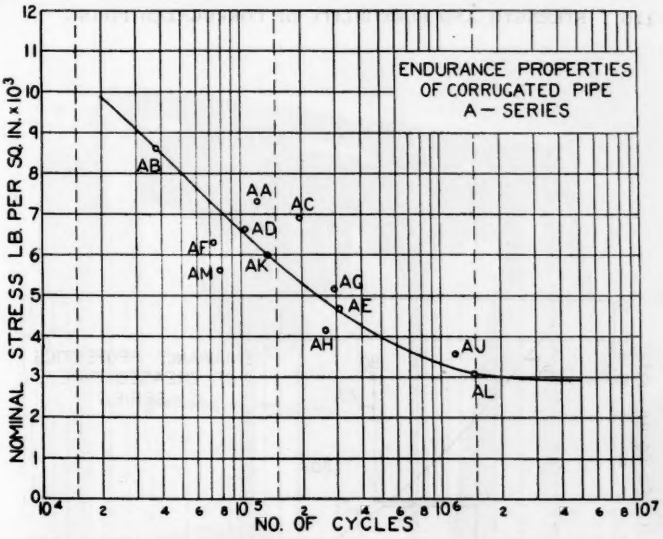


PLATE XIX—ENDURANCE PROPERTIES OF CORRUGATED PIPE, SERIES A.

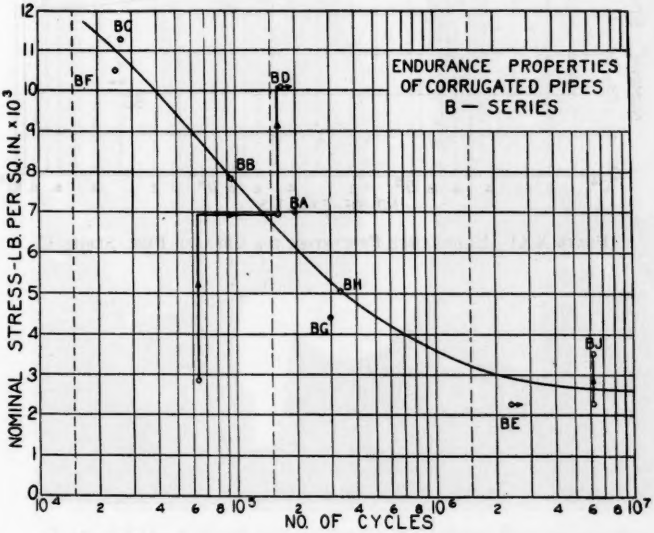


PLATE XX—ENDURANCE PROPERTIES OF CORRUGATED PIPE, SERIES B.

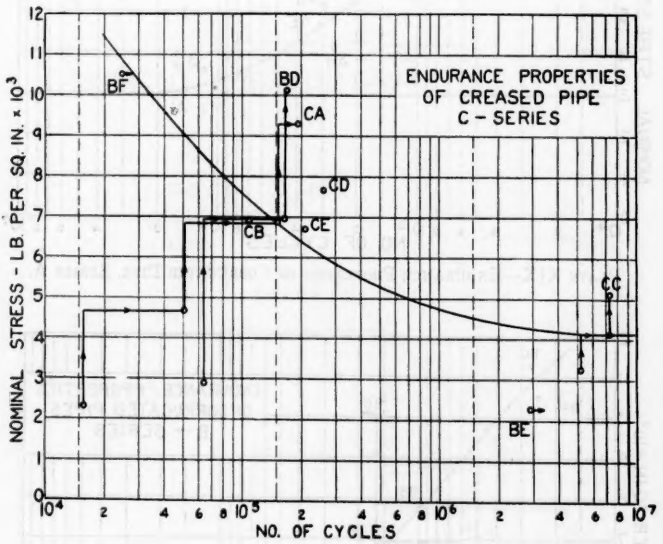


PLATE XXI—ENDURANCE PROPERTIES FOR CREASED PIPE, SERIES C.

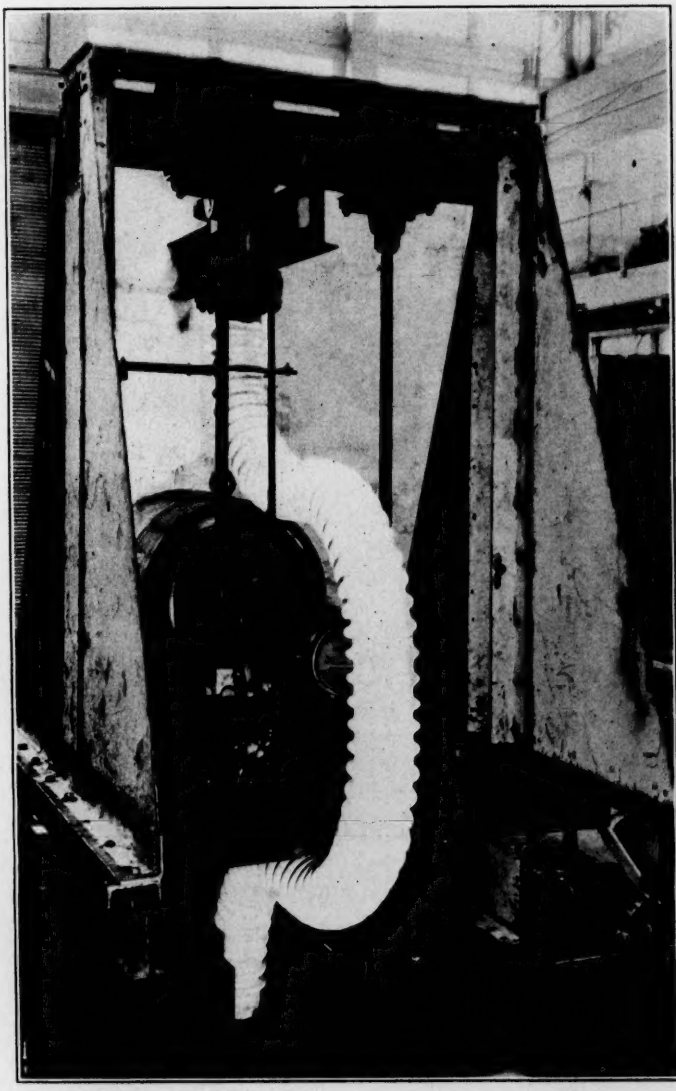


PLATE XXII—A CORRUGATED EXPANSION U-BEND ERECTED FOR FIXED-END TEST IN A SPECIFIC STRAIN TYPE FATIGUE TESTING MACHINE.

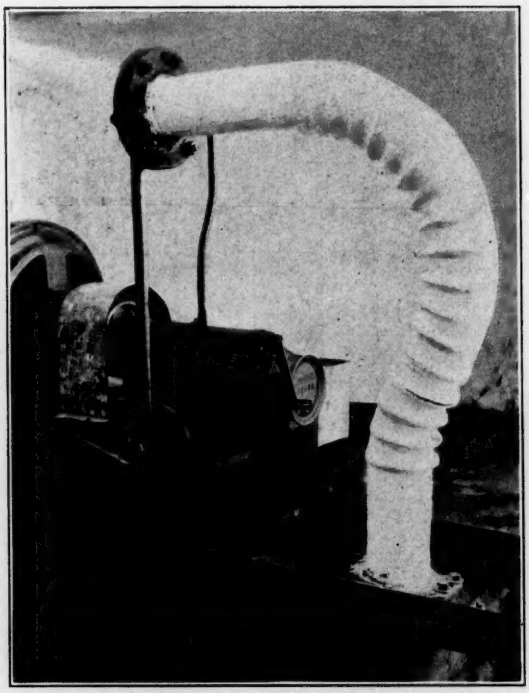


PLATE XXIII—THE SINGLE OFFSET QUARTER BEND AS ERECTED FOR TEST IN A SPECIFIC STRAIN TYPE FATIGUE TESTING MACHINE.

Note the crack, as a result of the fatigue test, which appears in the fourth corrugation from the bottom of the pipe.

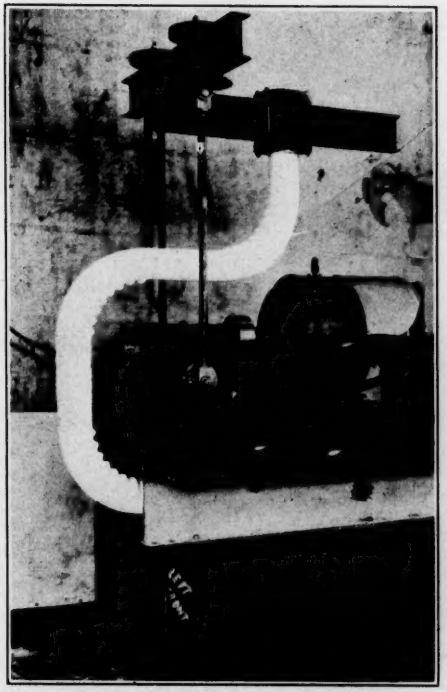


PLATE XXIV—A CREASED EXPANSION U-BEND ERECTED FOR TEST IN A SPECIFIC STRAIN TYPE FATIGUE TESTING MACHINE.

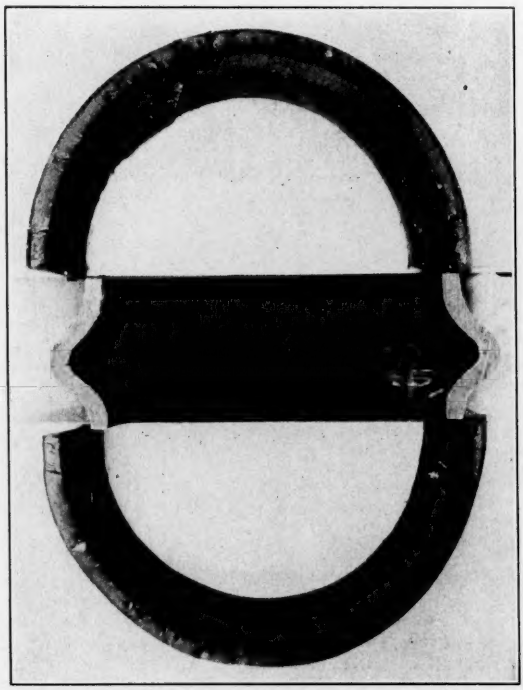


PLATE XXV—A FATIGUE FAILURE IN CORRUGATED PIPE, SERIES B.

This failure occurred in the rotating beam testing machine. The crack developed through the corrugation for about half the circumference. The pipe was sectioned by cutting out the failed corrugation and then by cutting through this corrugation along a pipe diameter. The two failed sections were laid back for the purpose of photographing. The appearance of the fracture indicates that the crack originated from the inner pipe surface.

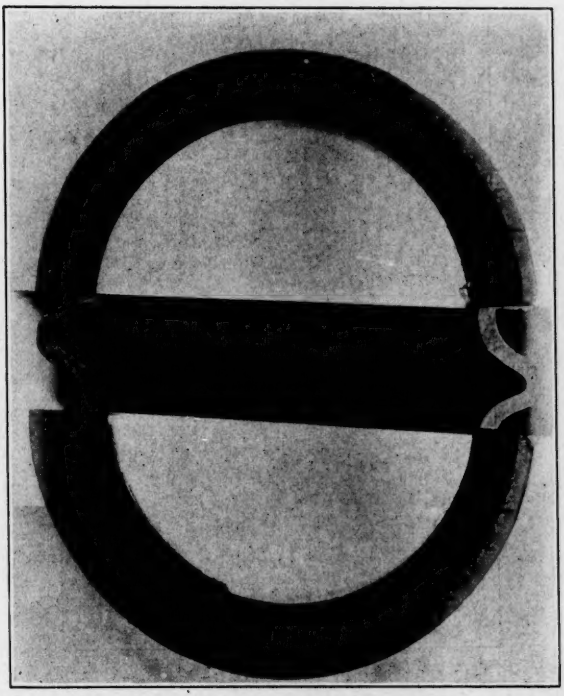


PLATE XXVI—A FATIGUE FAILURE IN CORRUGATED PIPE, SERIES A.

This failure occurred in a specific strain testing machine. The nature of the failure is similar to that occurring in the rotating beam machine as shown in Plate XXV. Here also the crack has progressed from the inner pipe wall to the outer surface.

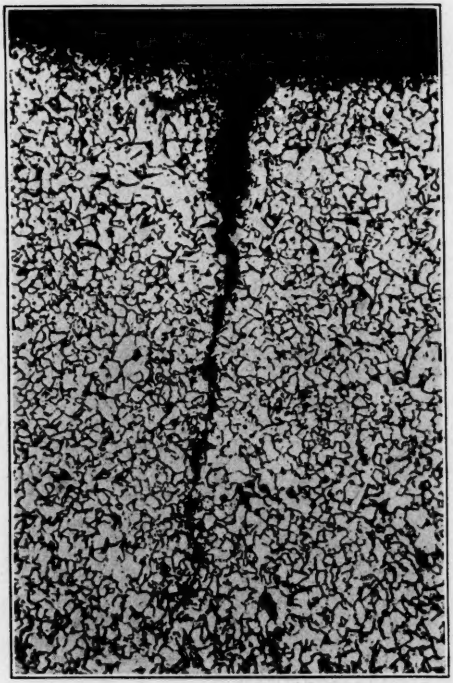


PLATE XXVII—PHOTOMICROGRAPH OF FATIGUE FAILURE.
(Magnification x100.)

This specimen was removed from piece A M, the single offset quarter bend, in the plane of the bend at the crest of the corrugation next above the fracture. The inner pipe surface is at the top of the photograph. The crack had not developed sufficiently to work entirely through the pipe wall.

PLATE XXVIII—A FATIGUE FRACTURE IN A CREASED PIPE.

This view is of a trough between two creases and shows the location and appearance of a typical failure in the creased pipe.

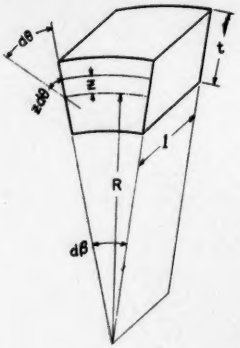


FIG 1

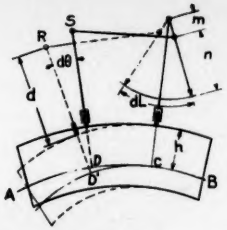


FIG. 2

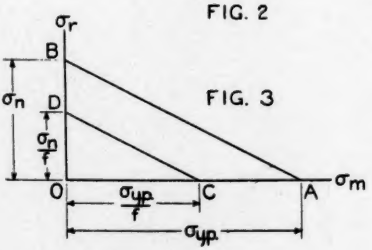
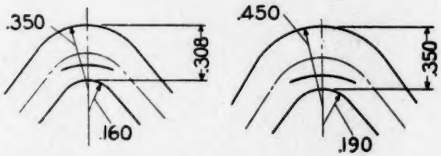


FIG. 3



SERIES-A SERIES-B

FIG. 4

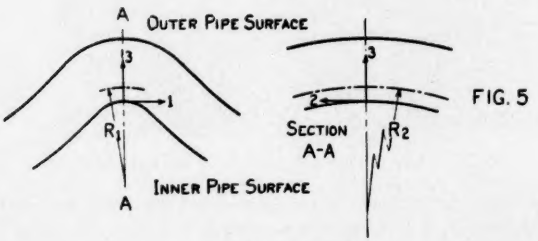


FIG. 5

PLATE XXIX.

FIG. 1—AN ELEMENT FROM A CURVED PLATE IN BENDING.

FIG. 2—A SCHEMATIC DIAGRAM OF THE ACTION OF THE BENDING STRAIN GAGE.

FIG. 3—TO ACCOMPANY SECTION 51, "THE METHOD OF DETERMINING SAFE WORKING STRESSES."

FIG. 4—THE SYSTEM OF AXES AND THE LOCATION OF R_1 AND R_2 FOR A CORRUGATED SECTION.

FIG. 5—TYPICAL CORRUGATION DIMENSIONS OBTAINED FROM AVERAGING MEASUREMENTS.

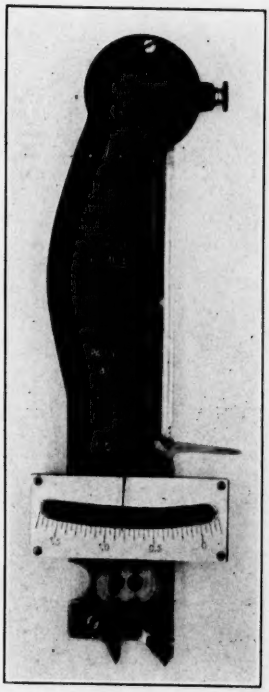


PLATE XXX—A VIEW OF THE HUGGENBERGER TENSOMETER.

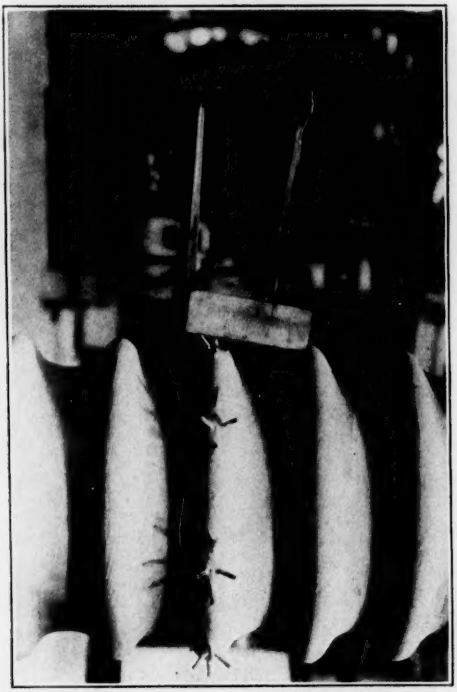


PLATE XXXI—THE BENDING STRAIN GAGE.

This view shows the gage applied to measure the longitudinal strain at the crest of a corrugation. The small rods for the base of the gage may be seen on the corrugation at several stations for measurement. The rods in the circumferential, or hoop, direction are set at a 1-inch gage length, while the ones in the longitudinal direction are set at ¼-inch. A calibration of this instrument showed it to have a magnification factor of 40 when applied to measure longitudinal strains and a factor of 45 when applied to measure hoop strains.

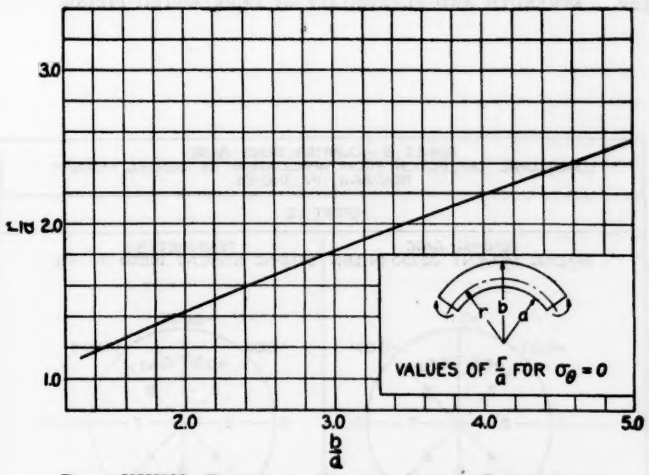


PLATE XXXII—POSITION OF NEUTRAL AXIS IN CURVED BARS.

The value of the ratio of the radii of curvatures of the neutral axis to the inner surface of a curved bar as a function of the value of the ratio of the radii of curvatures of the outer surface to the inner surface.

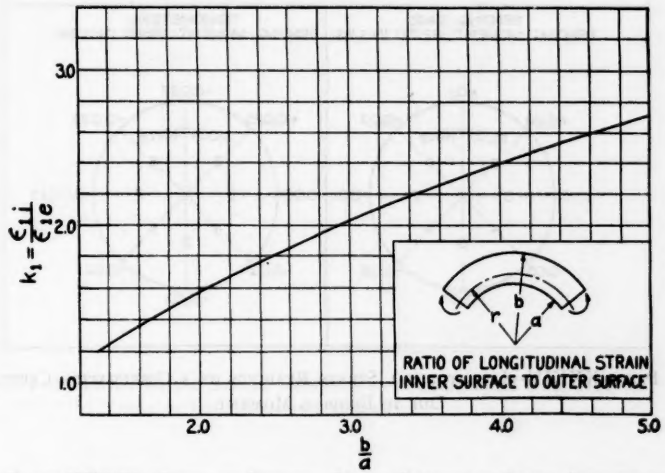


PLATE XXXIII—RATIO OF STRAINS, OUTER SURFACE TO INNER SURFACE IN CURVED BARS.

The absolute value of the ratio of the longitudinal strains at the outer surface to the inner surface of a curved bar in pure bending.

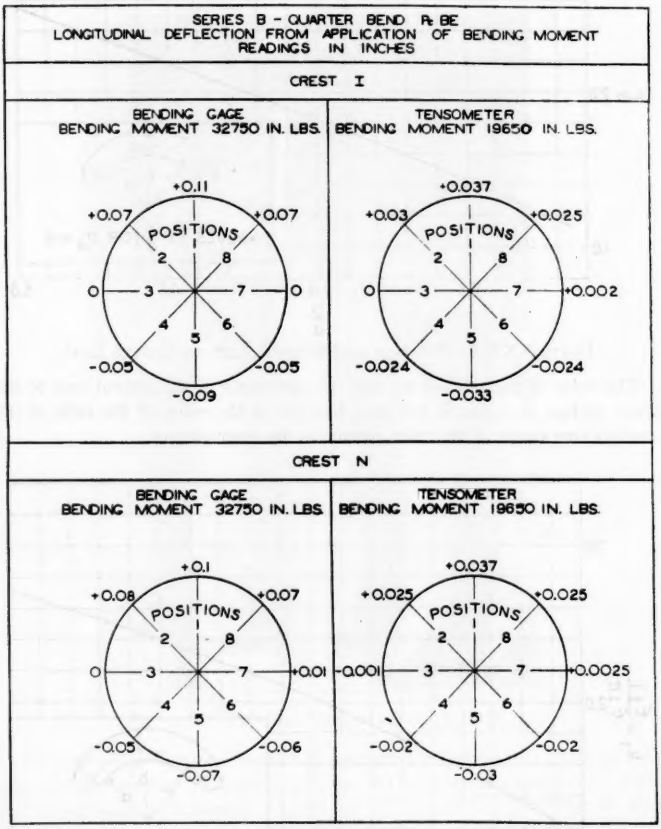


PLATE XXXIV—LONGITUDINAL STRAIN READINGS ON A CORRUGATION CREST, DUE TO BENDING MOMENT.

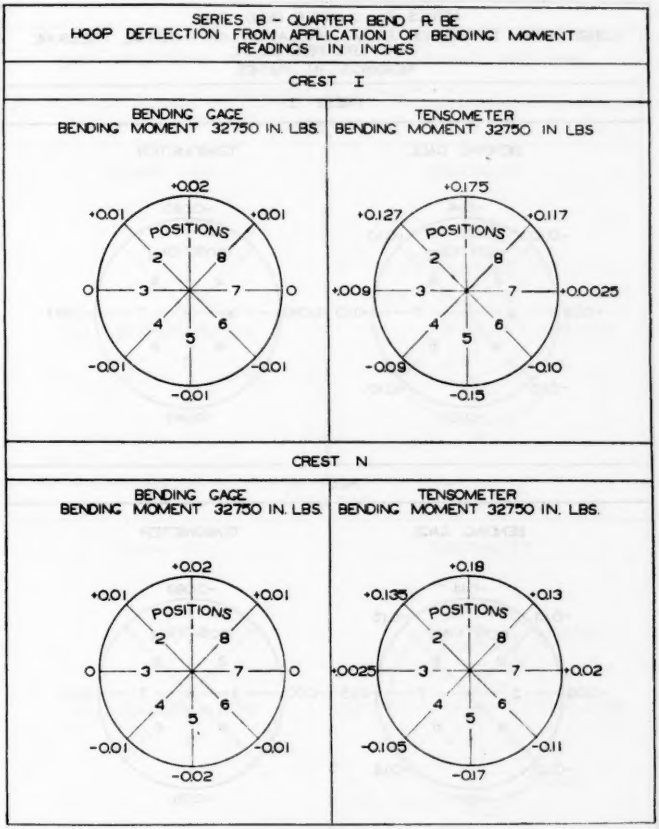


PLATE XXXV—HOOP STRAIN READINGS ON A CORRUGATED PIPE, DUE TO BENDING MOMENT.

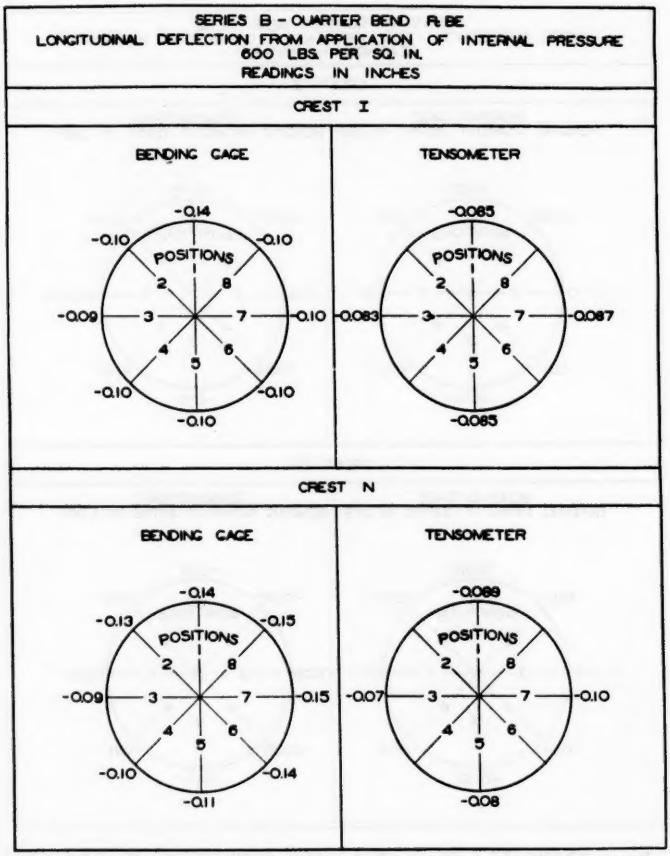


PLATE XXXVI—LONGITUDINAL STRAIN READINGS ON A CORRUGATED PIPE, DUE TO INTERNAL PRESSURE.

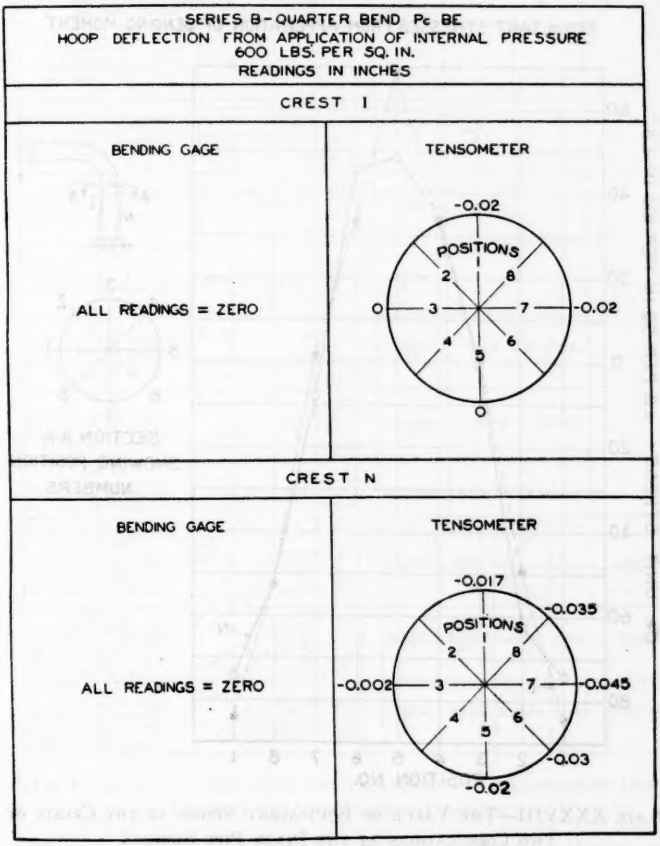


PLATE XXXVII—HOOP STRAIN READINGS ON A CORRUGATED PIPE, DUE TO INTERNAL PRESSURE.

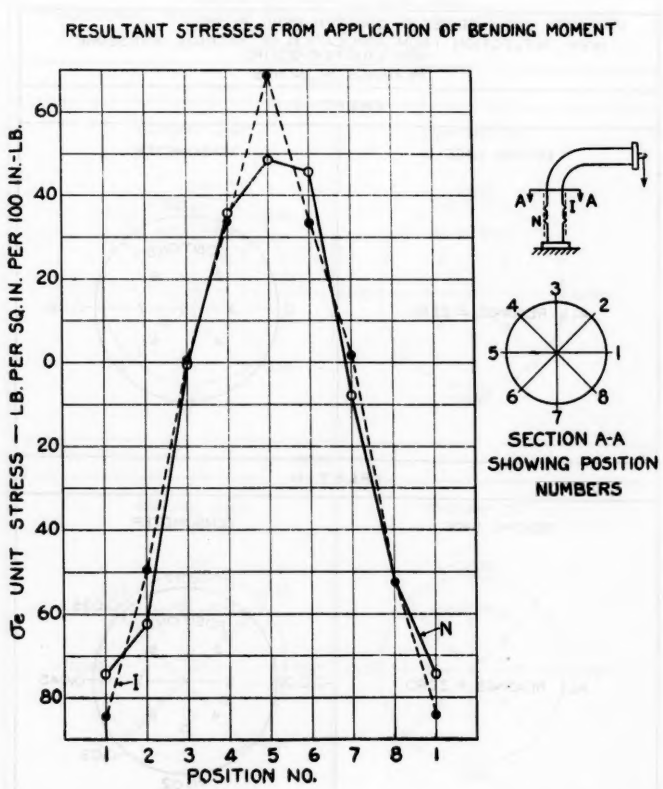
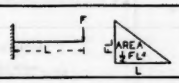
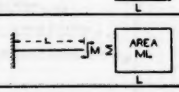
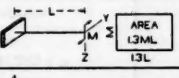
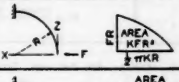
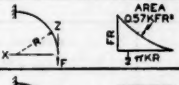
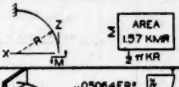


PLATE XXXVIII—THE VALUE OF EQUIVALENT STRESS AT THE CRESTS OF TWO CORRUGATIONS AT THE INNER PIPE SURFACE.

FROM "PIPING HANDBOOK" BY J H WALKER AND SABIN CROCKER SECOND EDITION		γ ROTATION	δ_x	δ_y
CASE I FIXED END OF A CANTILEVER ROTATATED THRU AN ANGLE γ			γL	
CASE II STRAIGHT CANTILEVER ACTED ON BY A TRANSVERSE FORCE F AT THE FREE END		$\frac{1}{2}\frac{FL^2}{ET}$	$\gamma\frac{2}{3}L$	
CASE III STRAIGHT CANTILEVER ACTED ON BY A MOMENT M AT THE FREE END		$\frac{ML}{ET}$	$\gamma\frac{1}{2}L$	
CASE IV STRAIGHT CANTILEVER ACTED ON BY A TORSIONAL MOMENT M		$\frac{1.3ML}{ET}$		
CASE V QUARTER BEND FIXED AT ONE END AND ACTED ON BY A RADIAL FORCE F IN PLANE OF BEND		$\frac{KFR^2}{ET}$	$\gamma 0.5R$	$\gamma 0.7854R$
CASE VI QUARTER BEND FIXED AT ONE END AND ACTED ON BY AN AXIAL FORCE F IN PLANE OF BEND		$\frac{0.57KFR^2}{ET}$	$\gamma 0.624R$	$\gamma 0.876R$
CASE VII QUARTER BEND FIXED AT ONE END AND ACTED ON BY A BENDING MOMENT M IN PLANE OF BEND		$\frac{1.570KMR}{ET}$	$\gamma 0.363R$	$\gamma 0.637R$

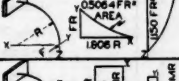
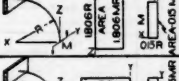
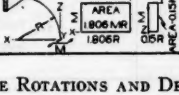
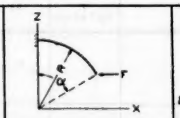
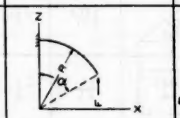
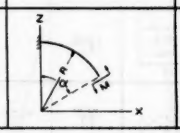
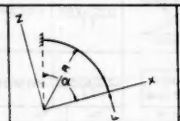
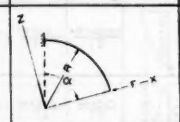
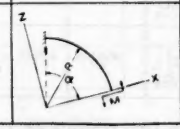
		γ_{xy}	γ_{zy}	δ_{xy}	δ_{zy}	δ_y
CASE VIII QUARTER BEND FIXED AT ONE END AND ACTED ON BY A FORCE F $\perp$ TO PLANE OF BEND		$\frac{.5064FR^2}{ET}$	$\frac{1.150FR^2}{ET}$	$0.734R\gamma_{xy}$	$0.754R\gamma_{zy}$	$\frac{2.983FR^3}{ET}$
CASE IX QUARTER BEND FIXED AT ONE END AND ACTED ON BY A MOMENT M IN A PLANE $\perp$ TO PLANE OF BEND		$\frac{.015MR}{ET}$	$\frac{1.806MR}{ET}$	$0.695R\gamma_{xy}$	$0.695R\gamma_{zy}$	$\frac{1.130MR^2}{ET}$
CASE X QUARTER BEND FIXED AT ONE END AND ACTED ON BY A TORQUE MOMENT M IN A PLANE $\perp$ TO FREE END OF BEND		$\frac{1.806MR}{ET}$	$\frac{.015MR}{ET}$	$0.306R\gamma_{xy}$	$0.306R\gamma_{zy}$	$\frac{.5064MR^2}{ET}$

PLATE XXXIX—THE ROTATIONS AND DEFLECTIONS OF A CANTILEVER AND A QUARTER BEND FOR VARIOUS LOADINGS.

These cases were taken from the "Piping Handbook" by Walker and Crocker.

COORDINATE AXES SELECTED SO THAT Z AXIS IS NORMAL TO THE ARC TANGENT AT THE FIXED END		
CASE XI FORCE AT FREE END ⊥ TO Z AXIS		MOMENT = $FR(1-\cos\alpha)$ $\gamma_F = \frac{KFR^2(\sin\alpha-\alpha\cos\alpha)}{EI}$ $\delta_x = \frac{KFR^3[\alpha(0.5+\cos^2\alpha)-1.5\sin\alpha\cos\alpha]}{EI}$ $\delta_z = \frac{KFR^3(0.5\sin^2\alpha-\alpha\sin\alpha\cos\alpha+\cos\alpha-\cos^2\alpha)}{EI}$
CASE XII FORCE AT FREE END ∥ TO Z AXIS		MOMENT = $FR\sin\alpha$ $\gamma_F = \frac{KFR^2(\alpha\sin\alpha+\cos\alpha-1)}{EI}$ $\delta_x = \frac{KFR^3(1.5\sin^2\alpha-\alpha\sin\alpha\cos\alpha+\cos\alpha-1)}{EI}$ $\delta_z = \frac{KFR^3[\alpha(0.5+\sin^2\alpha)+1.5\sin\alpha\cos\alpha-2\sin\alpha]}{EI}$
CASE XIII MOMENT AT FREE END		MOMENT = CONSTANT $\gamma_M = \frac{KMR\alpha}{EI}$ $\delta_x = \frac{KMR^2(\sin\alpha-\alpha\cos\alpha)}{EI}$ $\delta_z = \frac{KMR^2(\alpha\sin\alpha+\cos\alpha-1)}{EI}$
COORDINATE AXES SELECTED SO THAT X AXIS IS NORMAL TO THE ARC TANGENT AT THE FREE END		
CASE XIV FORCE AT FREE END ⊥ TO X AXIS		MOMENT = $FR(1-\cos\alpha)$ $\gamma_F = \frac{KFR^2(\alpha-\sin\alpha)}{EI}$ $\delta_x = \frac{KFR^3(1-\cos\alpha-0.5\sin^2\alpha)}{EI}$ $\delta_z = \frac{KFR^3(1.5\alpha-2\sin\alpha+0.5\sin\alpha\cos\alpha)}{EI}$
CASE XV FORCE AT FREE END ∥ TO X AXIS		MOMENT = $FR\sin\alpha$ $\gamma_F = \frac{KFR^2(1-\cos\alpha)}{EI}$ $\delta_x = \frac{KFR^3(0.5\alpha-0.5\sin\alpha\cos\alpha)}{EI}$ $\delta_z = \frac{KFR^3(1-\cos\alpha-0.5\sin^2\alpha)}{EI}$
CASE XVI MOMENT AT FREE END		MOMENT = CONSTANT $\gamma_M = \frac{KMR\alpha}{EI}$ $\delta_x = \frac{KMR^2(1-\cos\alpha)}{EI}$ $\delta_z = \frac{KMR^2(\alpha-\sin\alpha)}{EI}$

FROM A.S.M.E. TRANSACTIONS FSP-54-12

PLATE XI.—THE ROTATIONS AND DEFLECTIONS OF AN ARC OF LESS THAN 90 DEGREES UNDER VARIOUS LOADINGS.

These cases were taken from A.S.M.E. Transactions FSP-54-12 "Load Deflection Relations for Large Plain, Corrugated, and Creased Pipe Bends," by E. T. Cope and E. T. West.

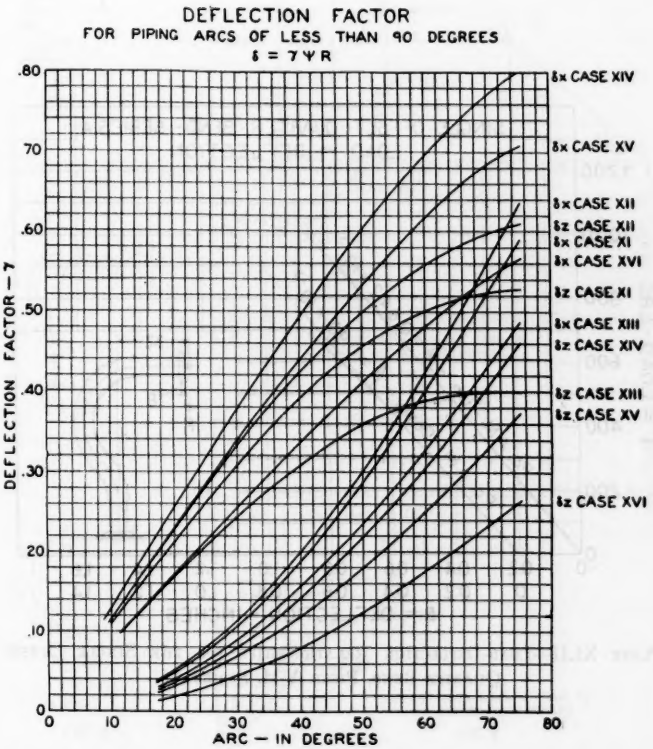


PLATE XLI—DEFLECTION FACTORS FOR AID IN COMPUTING DEFLECTIONS FOR ARCS LESS THAN 90 DEGREES.

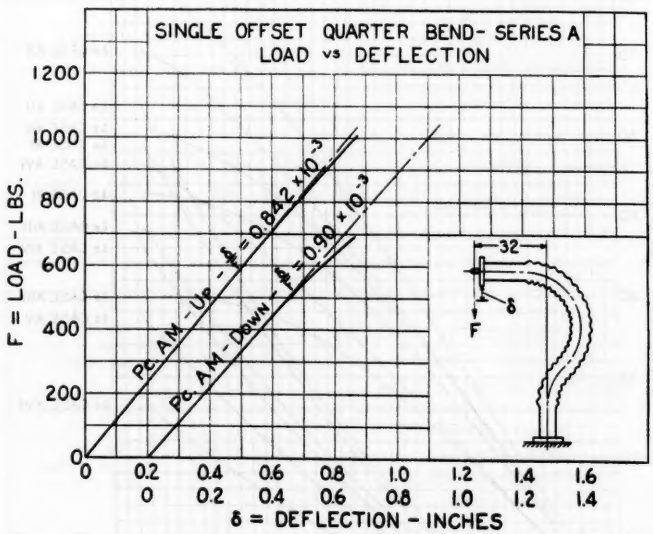


PLATE XLII—LOAD-DEFLECTION RELATIONSHIPS FOR THE SINGLE OFFSET QUARTER BEND, PIECE A M, SERIES A.

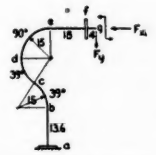


FIG 1

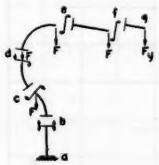
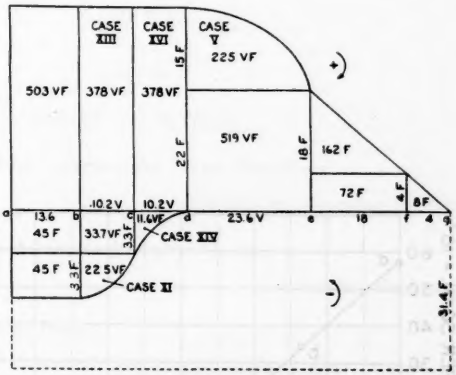


FIG. 2



MOMENT AREA DIAGRAM FROM F_y AND RESTRAINING MOMENT

FIG. 3

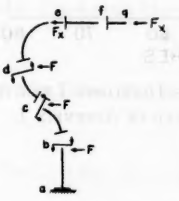
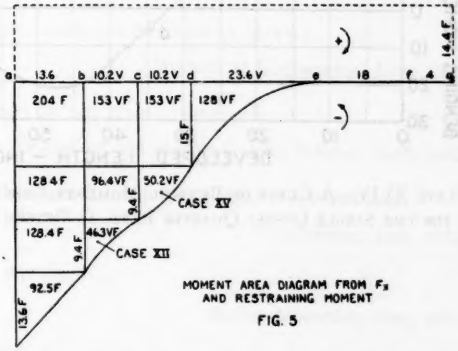


FIG 4



MOMENT AREA DIAGRAM FROM F_x AND RESTRAINING MOMENT

FIG. 5

PLATE XLIII—FIVE SKETCHES AND DIAGRAMS TO ACCOMPANY THE CALCULATIONS IN APPENDIX I.

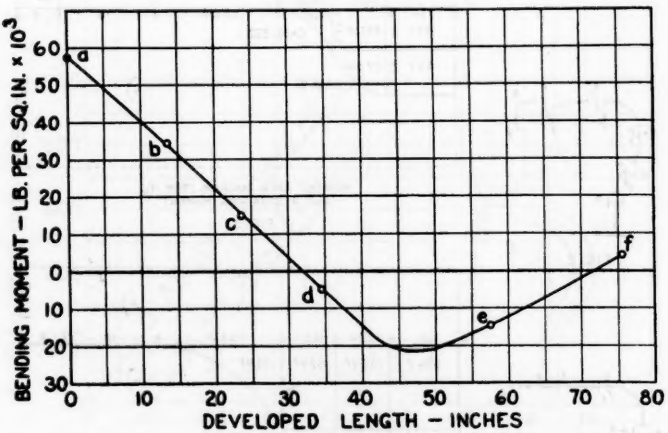


PLATE XLIV—A CURVE OF BENDING MOMENTS VERSUS DEVELOPED LENGTH FOR THE SINGLE OFFSET QUARTER BEND, AS DETERMINED IN APPENDIX I.

NOTES.

INDEX TO NOTES.

High-Pressure and High-Temperature Steam Pipe Work.
Engineering, December 16, 1932.

Stress Relief of Fusion-Welded Pressure Vessels.
Combustion, May, 1935.

The Trend of Destroyer Design.
The Shipbuilder and Marine Engine-Builder, April, 1935.

New Naval Ships.
The Engineer, 24 May, 1935.

Influence of Protective Layers on the Life of Metals.
Mechanical Engineering, June, 1935.

The Corrosion Problems of the Naval Architect.
The Shipbuilder and Marine Engine-Builder, April, 1935.

What Every Boiler Operator Should Know About Superheaters.
Combustion, June, 1935.

The Largest Marine Oil Engine.
British Motorship, June, 1935.

Steam Propulsion—What of the Future?
The Marine Engineer, April, 1935.

Burning of Cracked-Residue Fuel Oils.
Mechanical Engineering, April, 1935.

"Why the United States Requires a Merchant Marine."
Marine Progress, July, 1935.

HIGH-PRESSURE AND HIGH-TEMPERATURE STEAM PIPE WORK.*

By J. Arthur Aiton, J.P., C.B.E.

The design of the means for the conveyance of high-pressure and high-temperature steam from the superheater to the power unit is by no means simple, for the pipe-work is subjected to stresses due to pressure and expansion to which no other parts of the installation are liable, and these must be absorbed as far as possible within the pipe-work. Pipe-work must be considered by itself, and methods which may be quite satisfactory in boiler practice may not be so in piping, as, for instance, riveting on steam piping and vessels. Electrically welded steam drums are being introduced on boilers, but if satisfactory there, it does not follow that this method of manufacture will be satisfactory for steam vessels in pipe-work. Again, a boilermaker's remedy for pressure is thickness, whereas the pipe designer is constantly fighting against excessive thickness, for an excessively thick pipe is not necessarily a strong pipe; it is very stiff, and is therefore subject to heavy expansion stresses, which may quite outweigh any advantage gained from low bursting stress.

Pipe-work must be approached from three angles: first, the material from which the tube must be made, and the method of manufacture of the tube as it affects the pipe; then the pipe and its method of manufacture; and then the arrangement of the piping. One thing dominates all three, viz., the temperature of the steam. As it rises, the expansion of the pipes rises and the strength of the pipes falls, necessitating increase in thickness, consequently in stiffness; thus, the pipes are less able to take up the movements due to expansion, although these have been increased in magnitude. So far, mild steel, 0.15 to 0.19 per cent carbon, 24 to 29 tons, is the only material from which high-pressure and high-temperature pipes have been made in this country, and, owing to the action of this material under high temperature, pipes have already become excessively thick.

Of late years, with the advent of these high temperatures, engineers have been forced to take serious notice of creep in steel. Not very much is known of it, and the engineer has been forced to be very conservative in his treatment of it. Creep is the term which has been applied to the slow and continuous stretch which takes place in steel under a steady load. At atmospheric temperature it is either absent or so slow as to be negligible, but as temperature is raised, creep becomes serious. Stress, temperature, and time form the equation which determines the amount of creep. To the engineer, permissible stress is the unknown; he can state the quantity of creep allowable, the time during which it will take place and the temperature under which it shall take place, but he must refer to the metallurgist for the permissible stress. If the maximum life of piping be taken at twenty years, 175,200 hours, it will be safe to compute the maximum time during these twenty years that the piping will be under full temperature and pressure to be 100,000 hours

The allowable amount of thinning due to stretch in 100,000 hours may be taken at 1 per cent, *i.e.*, one-tenth of 1,000,000th inch per inch per hour, or about one thousandth per inch per average year. Temperature is fixed by the design of the station, and may be taken at the maximum of 900 degrees F. The question, then, for the metallurgist is: what stress will produce a creep of one-tenth of 1,000,000th inch per inch per hour at 900 degrees F.; and on his answer depends the design of the piping. These calculations are based on

* Paper read before the Electrical Power Engineers' Association, London Local Group, on Tuesday, October 25, 1932. Abridged.

the assumption that the rate of creep is constant. This is not correct, for it is known that during the first few hours the rate of creep is comparatively rapid, but that it settles down to a fairly regular rate, which, however, decreases slowly at first, and more rapidly as the years go on. It is this decrease of the rate of creep in the efflux of time which is not yet determined, but as it is all favorable, it may be neglected at present.

In the light of available data today, the metallurgist informs us that at 900 degrees F. the stress to produce conditions specified is 3 tons, but at 1000 degrees F. the stress falls to 1½ tons in 0.17 per cent carbon steel, which means that this material is not suitable for temperatures over 900 degrees F. For 0.17 per cent carbon steel working to 60 per cent of the stress to produce the creep rate specified, the following are suitable maximum working stresses: 900 degrees, 4500 pounds; 850 degrees, 5500 pounds; 800 degrees, 6500 pounds; 750 degrees, 7500 pounds; 700 degrees, 8500 pounds. These stresses are below the stresses which will produce a creep rate of one-hundredth of 1,000,000th inch per inch per hour, or one-tenth of 1 per cent during the life of the pipe.

Having fixed the limit working stresses, their effects on practice can now be considered, assuming two stations, the one working at 1200 pounds pressure, 700 degrees, and the other at 750 pounds, 900 degrees. In the first, with a working stress of 8500 pounds, a ten-inch bore pipe would be ¾ inch thick, and in the second, with a working stress of 4500 pounds, a similar pipe would be ⅞ inch thick. If, however, in the first instance, that is at 1200-pound pressure, the temperature is gradually raised, the thickness would be: at 750 degrees, 13/16 inch; 800 degrees, 15/16 inch; 850 degrees, 1⅛ inches; 900 degrees, 1⅜ inches. Excessively thick tubes present so many difficulties in manufacture that it would not be easy to work at 750 degrees with this pressure, and would not be practical to work at any higher temperature; in other words, we have reached a limit beyond which we cannot go with mild steel—0.15 to 0.19 per cent carbon, 0.6 per cent manganese, 0.04 per cent silicon and 0.04 per cent phosphorus.

If temperature is not to remain low at high pressure, or pressure to remain low at high temperature, an alloy steel must be found which will give better results under high-temperature conditions. There are many alloy steels already on the market which do this, but they require special heat treatment. Steel for pipe manufacture has to be capable of being worked at high temperatures in the manufacturing processes without requiring elaborate heat-treatment afterwards. Alloy-steel stud bolts are regularly used, provided with a nut at each end in order that the material shall not be worked under heat after leaving the steelmakers' works.

The maker of solid-drawn hot-finish tubes requires a tolerance of 12½ per cent up or down, which means that the pipe must be specified 12½ per cent thicker than the minimum thickness required, and may come out 12½ per cent thicker than that specification. This means that a pipe required to be not less than ⅞ inch thick must be ordered 1 inch thick, and must be accepted if it comes 1⅛ inches thick. This increase of thickness does not affect the strength of the pipe, for as the stress in the metal due to expansion goes up, the stress due to internal pressure comes down, but it does increase the thrust on the fixed points to which the pipes connect, and may be serious to them.

A custom is growing up of specifying that pipes shall be made to the actual bore with small tolerances. This tends to thicken the pipe wall still further, as these pipes are made to the outside diameter. Take the case of 10-inch pipe ⅞ inch thick being ordered 1 inch thick. The outside diameter may be 12¼ inches; the nearest standard is 12½ inches. If the actual 10-inch bore must be produced, the pipe must be made 1¼ inches thick to

conform to the bore, or nearly double the necessary thickness, whereas if the exact bore had not to be maintained the pipe can be made to the specified thickness and the bore would vary between 10¼ inches and 10¾ inches. If the bore must be maintained, the question of making these pipes from cold-finish tubes must be considered, as these tubes are made on a mandrel to actual bore and the only tolerance required in thickness is 10 per cent up. Cold-finish tube is much more expensive than hot-finish, but when all the circumstances are considered, and if the makers of cold-drawn tubes will bring their prices down, it may well prove to be the cheaper in the end.

As to the method of joining the pipes together, it may at once be stated that with modern conditions the only satisfactory method is by welding. The welding is a definite seal which, once made tight, should continue in that condition indefinitely. The best joint is that in which the pipes are welded together with ends butting, after which a heavy sleeve is passed over the joint and welded to both of the pipes, but it is not easy to erect and weld on site, it can only be broken with difficulty, and it cannot be made to connect to valves and cast fittings. The objection to dependence on the weld alone and unsupported is not based on practice, as these joints have proved quite satisfactory in use at high pressures and temperatures over a number of years, but to meet it joints have been put forward combining screwed couplings and welding, and bayonet couplings and welding. They are subject to the objections above mentioned.

There is also the type in which flanges and welding are combined. Two of these have been used to a considerable extent and may be said to occupy the field. They are known as the "Sarlun" joint and as the "Corwel" joint. The former has been used fairly freely in America and to a much lesser extent in this country and on the Continent; the latter is British, and is being used in most modern power stations in this country, to a small extent on the Continent, and is not yet being used in America. Both joints have collars formed on the ends of the pipes, integral with the pipe, backed by heavy loose flanges which are bolted together and take all mechanical and expansion stresses. The joint is made steam-tight by a weld round the junction of the two collars. Both these joints are made to connect to valves or fittings, and blanks. They can also both be broken easily; the "Sarlun" not many times, but the "Corwel" has been broken and remade half a dozen times. As stiffness is the desideratum in the flange, a higher carbon steel can be used.

The bolts are of nickel-chrome-molybdenum steel alloy; ultimate stress, cold, 69 tons. These are screwed fine thread all the way along, and have a small bright nut at each end, all to B.S.T. No. 191–1924. These stud bolts in a 16-inch high-pressure, high-temperature "Corwel" flange are 1¾ inches diameter, 15⅜ inches long, and weigh, with nuts, 12¼ pounds each; as there are 20 of the bolts in each joint, the bolt weight is 2 hundredweight 1 qr. per pipe.

The bending of thick pipes requires serious consideration. With a 10-inch pipe ⅜ inch thick and a radius of 40 inches, four times the bore, which was a standard a short while ago, the neutral axis of the bend is normally at the center line of the pipe; the outer wall will thin 10 per cent and the inner wall will thicken 10 per cent. This thinning was not serious in the days when the pipes were worked well within the maximum permissible stress, and, further, it did not always take place, as with careful bending the neutral axis could be shifted out past the center line of the pipe and nearer the outer wall, so reducing the thinning to a negligible amount. With thicker pipes, however—say a 10-inch pipe ⅞ inch thick, 40 inches radius—the

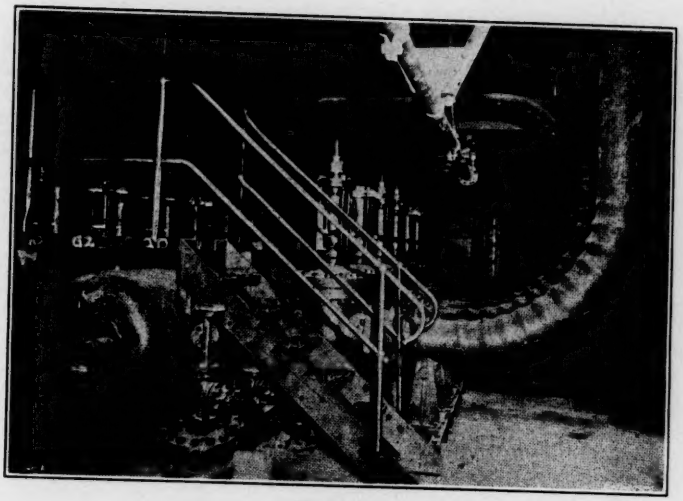


FIGURE 1.

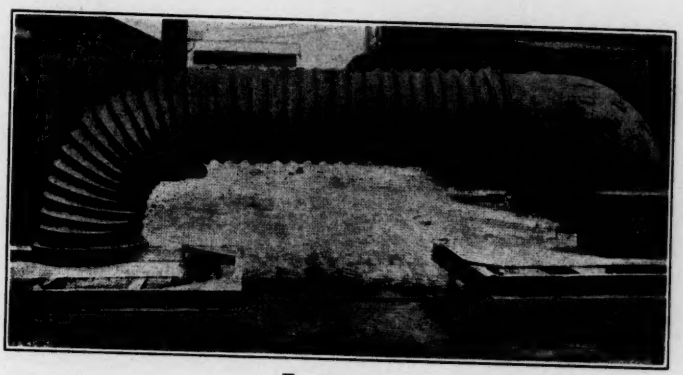


FIGURE 2.

neutral axis is actually in the inner wall of the pipe, *i.e.*, the inner wall of the bend does not decrease in length nor increase in thickness, but the outer wall has to increase the total variation in length and decreases in thickness by 30 per cent, which means that that pipe cannot be bent to so small a radius. An easy calculation shows that this pipe must be bent to a minimum radius of 11 times the bore in order that the outer wall shall not be thinned more than 10 per cent.

From this it follows that a bend made in the ordinary way from 10-inch pipe ⅞ inch thick must have a radius of not less than 10 feet. From the point of view of passing steam such a large radius bend is quite satisfactory, and if there is sufficient space to accommodate it nothing can be said against it. But in all power stations the pipe bends serve the dual purpose of passing the steam and providing some of the flexibility necessary to allow the movements due to variations in temperature to take place. It is an axiom that the larger the radius of the bend the lower will be the flexibility of the pipe line into which it is incorporated, and it is for this reason that some other method of bending has had to be found by which the shortening of the inner wall of the bend can be provided for without the necessity of increasing its thickness, thereby moving the neutral axis back to the center line of the bend, or even beyond that nearer to the outer wall. This method consists of raising a series of creases on the inner wall which extend circumferentially and vanish out at about two-thirds of the circumference of the pipe. These bends are known as creased bends; they are more flexible than bends of plain pipe made to the same radius, but we do not take this into account in our calculations. Their principal value is that they enable a bend to be made in thick pipe to a much smaller radius than can be made in plain pipe without damage. In comparing plain bends with creased and other special bends, a small radius-creased bend should be compared with the smallest radius bend that it is possible to make from plain pipe, and not one of the same radius.

Furthermore, creasing enables a bend to be made in larger bore lap-welded pipe, which bend could not be made by any ordinary method in plain pipe, as pieces would have to be cut out and the pipe welded up. These pipes are used for reheating steam at intermediate pressures from 100 pounds to 150 pounds, and the steam is reheated to 800 degrees. The pipes have been made up to 30-inch bore, ⅜ inch thick, but could be made up to 40-inch bore, ½ inch thick. There is a third method of making bends, by raising a series of parallel corrugations round the pipe in the straight and then bending it; in this process the neutral axis remains at or near the center of the pipe. The variation in the length of the outer and of the inner wall is taken up in the corrugations, those on the outer wall straightening out somewhat and those on the inner wall closing up. These corrugated bends can be made to a much smaller radius even than creased bends, but their chief advantage is that they in themselves are very much more flexible than either plain or creased bends, and straight pipes can be made to help the flexibility very much by being corrugated at their ends for some distance. A group of creased pipes is shown in Figure 1, and a corrugated pipe in Figure 2.

Considerable care has to be taken in the manufacture of creased and of corrugated pipes: first of all, the tubes must be of good quality and well made, as surface defects are intensified and brought into view and laminations or other hidden defects betray themselves. Any variations in the thickness of the wall, either circumferentially or longitudinally, are shown in the creases and in the corrugations, and if they exceed quite moderate dimensions they render these processes impossible. But even the best pipes may be

damaged in creasing and in corrugating by the wrong form of crease or corrugation, which must vary according to the conditions which the pipe is to meet, the thickness of the pipe, and the radius of the bend. They can be damaged by bad heating—excessive heating may burn the outside of the pipe and incorrect heating will produce many small cracks on the inside of the corrugation, any one of which may develop into a fracture under the stress of working conditions. Incorrect design and careless treatment in bending may produce collapsed creases or corrugations, or corrugations of varying pitch and varying heights. All these defects in their advanced stages can be detected by any inspector, but it needs care and experience to detect them in their early stages, and they must be detected in their early stages, otherwise they may go into working and develop into fractures. Special apparatus for internal inspection is needed if the tops of the corrugations are to be examined; it is in the tops of the corrugations that these defects are to be found. All creased and corrugated bends must be annealed, and plain bends ought to be.

The question of dealing with expansion in modern pipe-work is most important. Each case must be considered on its merits, and very careful consideration backed by experience is necessary if serious trouble is to be avoided. With two bends of equal length of leg but with different radii of bend, the one with the smaller radius will be definitely more flexible than the one with the larger radius. Given two bends of the same pipe to the particulars given in Figure 3, radius R and 3 R, length of straight A C = 3½ R, A C = 1½ R, respectively; if the bends are assumed to be rigid, then the whole of the movement which the bends must take up must be got by springing the straight pipe, and as the short straight on the long-radius bend has to spring through a much larger angle than the long straight on the short-radius bend, the long-radius bend must be stiffer than the short-radius bend. Bearing this in mind, the particulars of six bends, shown in Figure 4, have been worked out and are given in the accompanying table. All the bends are 10-inch bore pipe, ⅞ inch thick, and the legs are 15 feet. The deflection in each case is 0.286 inch, and thrust and stress are worked out for this, the flange being kept parallel.

The fall both in thrust and stress between No. 1 and No. 2 is noteworthy, and is due to the difference in the length of straight. The further fall between No. 2 and No. 3 is due to the flexibility of the corrugated bend, which reduces the angle through which the straight has to be sprung. To reduce the stiffness in No. 2 and No. 3, it is necessary to introduce some flexibility on the straight legs; this has been done by corrugating the end of each for a length of R, as shown on No. 4 and No. 5. In No. 4 the maximum stress has been removed from A, near the flange, to C, near the bend, and considerably reduced even well below No. 3, the corrugated bend. In No. 5 the maximum stress is at A and has been reduced well below all the others, but is still too high, as the maximum stress due to expansion which can be permitted is 2250 pounds, half the total at this temperature, the other half being taken up by the internal pressure. Bend No. 6 shows the smallest arrangement which would give the necessary movement. The expansion of this bend itself at 900 degrees is 1.13 inches. By taking up three-fourths of this cold, the bend would just take up its own expansion, and the thrust hot on the connections would be 1190 pounds, and the stress in the metal at A 1700 pounds, with a movement of 0.286 inch; the thrust cold would be 4740 pounds, and the stress at A 6740 pounds, with a movement of 0.844 inch. These bends are simple, but they serve to show the effect of the combination of corrugated straights and corrugated or creased bends; such combinations, though varied in practice, are all governed by

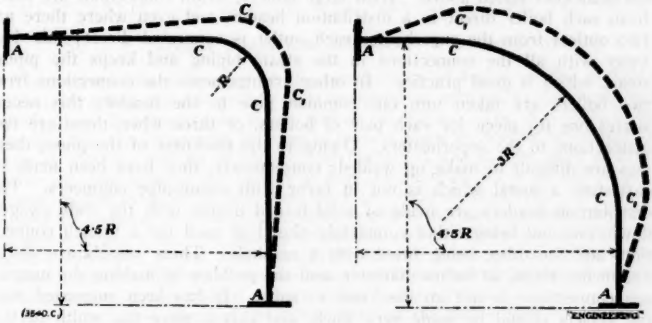


FIG. 3. DISTORTION OF THE PIPE BEND.

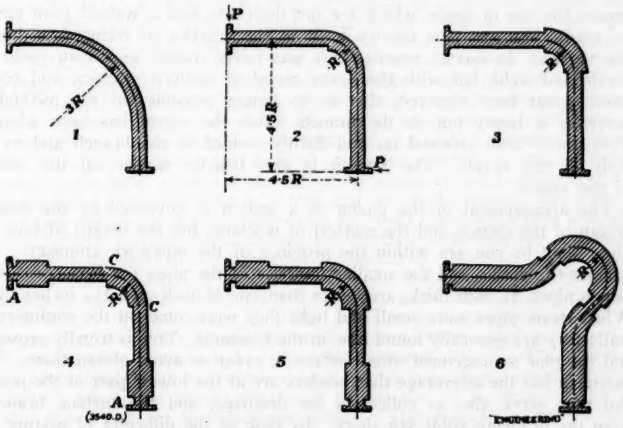


FIG. 4. COMPARATIVE FLEXIBILITY OF PIPE BENDS.

Conditions assumed :—

Working pressure = 700 lb. per square inch. Total temperature = 900 deg. F.
Pipe, 10 in. in internal diameter, $\frac{1}{4}$ in. thick. Radius = 40 in.
Allowable bending stress = 2,250 lb. per square inch.
Expansion on 15 ft. = 1·13 in. Temperature range = 850 deg. F.

Movement under cold strain = 0·844 in. Movement under hot strain = 0·286 in.

Corrugated pipe indicated by increased thickness both sides. Creased pipe indicated by increased thickness one side only.

Bend No.	1	2	3	4	5	6
Thrust, lb.	14,980	6,070	4,400	3,480	2,345	1,190
Stress, lb. per square inch	13,640	7,250	6,140	3,420	2,505	1,700

the principles stated above. With large boilers, steam connections are taken from each boiler direct to a distribution header, and even where there are two outlets from the superheater each outlet is connected direct; this does away with all the connections in the steam piping and keeps the piping small, which is good practice. In other arrangements the connections from two boilers are taken into one common pipe to the header; this necessitates one tee piece for each pair of boilers, or three when there are two connections to the superheaters. Owing to the thickness of the pipes, these tees are difficult to make up welded; consequently, they have been made of cast-steel, a metal which is not in favor with steam-pipe engineers. The distribution headers are made of solid-forged drums with the ends swaged down, one end being either completely closed or used for a branch connection, and the other being fitted with a manhole. These vessels are above 2½ inches thick, 30 inches diameter, and the problem of making the numerous connections is not an easy one to solve. It has been suggested that the vessels should be made very thick, and that a piece the width of the diameter of the largest flange should be planed off to which the flanges could be studded. There are many objections to this; it is very expensive, it means the use of studs, which are not desirable, and a welded joint cannot be used. We are thus thrown back to the practice of fixing branches to the vessels. In former practice this was fairly simple, as a man could get inside and weld, but with the heavy metal of modern practice, and consequent great heat required, this is no longer possible, so the method of screwing a heavy nut to the branch inside the vessel has been adopted. This nut is both screwed on and lightly welded to the branch and to the body of the vessel. The branch is also heavily welded on the outside of the vessel.

The arrangement of the piping in a station is governed by the general design of the station and the method of working, but the details of how the pipes shall be run are within the province of the pipework engineer. The simpler the design and the smaller the bore of the pipes the better, *e.g.*, two 9-inch pipes, ⅝ inch thick, are better than one 16-inch pipe 1⅛ inches thick. When steam pipes were small and light they were hung on the engine-room wall; they are generally found now in the basement. This is usually crowded, and the pipe arrangement often suffers in order to avoid obstructions. The basement has the advantage that headers are at the lowest part of the piping, and thus serve also as collectors for drainage, and the turbine branches from the drainage point are short. In view of the difficulty of getting the piping into the basement and its very restricted head-room for bends, an arrangement is coming in for building an intermediate floor on the boiler-house side of the engine-room wall at a level about half-way between the level of the boiler stop valves and the turbine connections. The headers and all cross-connecting pipes are located on this floor, which can be made large enough to allow ample room, not only for the piping but also for feed pumps and feed tanks. The turbine branches are led through into the engine room and down the wall into the basement, thence up to the turbine, arrangements for draining through a trap being provided at the lowest point on the branch piping. This makes a very good arrangement. The piping, while out of the way, is not out of sight, and should any work have to be done to the valves or other accessories, there is ample room to carry it out with convenience and expedition.—"Engineering," Dec. 16, 1932.

STRESS RELIEF OF FUSION-WELDED PRESSURE VESSELS.*

By D. S. Jacobus, President, American Welding Society.

There is no subject bearing on the construction of fusion-welded pressure vessels, on which there is such a wide variation of opinion, as the conditions which make it necessary to stress relieve the joints. The A.S.M.E. Code requires that the stresses be relieved in certain cases by heating uniformly to at least 1100 F., and up to 1200 F., or higher, if this can be done without distortion, and holding the vessels at that temperature for a period proportioned on the basis of at least one hour per inch of thickness. The Code covers the welding of plain carbon steels in which the carbon content does not exceed 0.35 per cent. All welded drums for power boilers and all pressure vessels employed in the most dangerous service are required to be stress relieved, but the Unfired Pressure Vessel Code sanctions the use of quite a wide range of fusion-welded vessels without stress relief.

The temperature specified for stress relieving the welds does not refine the grain structure. In vessels made of alloy steel or in which the carbon limit exceeds that specified in the Code, it is necessary in many cases to refine the grain adjacent to welding by heating the vessels to above the critical temperature after welding. The remarks which follow apply to the stress relief of vessels constructed of plain carbon steel in accordance with the rules of the A.S.M.E. Code where the carbon content does not exceed 0.35 per cent and may not apply to vessels constructed of other grades of steel.

The A.S.M.E. Unfired Pressure Vessel Code requires unfired pressure vessels, which are to be used under the most exacting service conditions, to be stress relieved, and others for a limited field of service to be stress relieved when the shell thickness for a given diameter exceeds a certain figure. Vessels of the latter class are required to be stress relieved where both the wall thickness is greater than 0.58 inch and the shell diameter less than 20 inches, and for all other wall thicknesses and shell diameters where the ratio of the diameter to the cube of the shell thickness is less than 100, and in all cases where the shell thickness is over 1½ inches.

The A.S.M.E. Unfired Pressure Vessel Code sanctions the use of unfired pressure vessels of the latter class for storing gases other than those of a lethal nature. Where water or other liquid is present in the vessel the pressure for a vessel that may be used without stress relief is limited to 400 pounds per square inch, and the temperature to 300 F. The object of these limitations where the vessels contain liquids is to insure safety for types of vessels where the energy developed in the case of an explosion by the heated liquid has led to disastrous results. The A.S.M.E. Code is more liberal in its stress relief requirements than any other Code that I know of. For example, the specifications of the Bureau of Engineering of the U. S. Navy call for stress relieving all fusion-welded pressure vessels, and the "Tentative Requirements of Lloyd's Register of Shipping for Fusion-Welded Vessels Intended for Land Purposes" call for heat treating all vessels subject to internal steam pressures above 50 pounds per square inch, and other vessels, depending on the service for which the vessels are intended.

One of the objects of the rule given in the A.S.M.E. Boiler Code was to exempt certain vessels from the requirements of stress relief where it would be impossible to stress relieve the vessels. The vessels in mind were those built in the field having large diameters, say 30 feet or more, where it would be impossible to stress relieve the vessels as a whole and where there would be more flexibility than in a smaller vessel of the same plate thickness.

* From address on Standards and Codes for Fusion Welding presented at Meeting of Local Sections of American Welding Society, American Society of Mechanical Engineers and American Society for Metals, Detroit, Mich., April 17, 1935.

The rule exempting certain unfired pressure vessels from stress relief which was finally embodied in the Unfired Pressure Vessel Code was prepared jointly by the American Welding Society and the A.S.M.E. Boiler Code Committee. It was published in "Mechanical Engineering," the official organ of the American Society of Mechanical Engineers, with a request that criticisms or suggestions be submitted to the Boiler Code Committee. No criticisms or suggestions were received before including the rule in the Unfired Pressure Vessel Code, whereas there has been considerable discussion as to the exactness of the rule since its inclusion and the Boiler Code Committee has been requested to provide a more liberal rule.

Many investigations are in progress and much has been published on the strains and the corresponding stresses due to welding. It has been shown that welding a seam in a pressure vessel may produce elongations in the plate adjacent and at right angles to the weld which will be two or more times as great as that required to produce a stress equal to the yield strength of the material. It has also been shown that the residual strains after the welding, both at right angles and parallel to the weld, may be in the neighborhood of those corresponding to the yield strength of the plate. It is well known that in fusion welding a pressure vessel it will at times rupture due to the strains produced by the welding before the application of the hydrostatic test in the shop, all of which shows that the strains due to the fusion welding are of great magnitude. The question arises as to the effect of these strains should the vessel be placed in service without stress relief.

There is a feeling on the part of some that if a vessel is placed in service after being subjected to the usual hydrostatic tests that the stresses will gradually diminish due to the creeping of the material and safety will result through the readjustment of the stresses, provided the welds are sound and the weld metal of as uniform and as good quality as the plate.

Vessels tested by subjecting them to pulsating pressures have ruptured in the metal of the plate parallel to the weld at lower stresses than correspond to the strength of the material. It has been shown that this action may come through the stresses adjacent to the weld produced by the welding, as these stresses added to the hoop stress produced by the hydrostatic pressure may correspond to or exceed the endurance limit of the material. For certain types of service, therefore, there would seem to be no question but that it is well to stress relieve all fusion-welded vessels.

A most important feature that must be considered in deciding whether a vessel should be stress relieved is the chance that the weld will not be as uniform and as strong as the parent plate. If portions of the welded joint are not thoroughly fused, experience has shown that intermittent stresses due to variations in temperature or pressure will lead to cracks starting at the unfused portions and extending so that danger may result. Stress raisers of the sort due to imperfections in the weld are something that must be given most serious consideration and in case of a doubt those responsible for the formulation of safety codes naturally lean toward conservative rules.

RAPID ADVANCES MAKE CAUTION ADVISABLE.

The results secured in long service are the best measure of safety. It is difficult, however, to formulate rules based on the results secured in service where the art is advancing as rapidly as in the case of fusion welding and where defects may be developed after many years of use. Again, the stresses depend in a great measure on the method used in welding, and many variables are involved. We must not be misled through the rapid advances that have been made in the last few years into thinking that we know all about fusion welding. As I have said on other occasions, I do not believe

we have proceeded over 25 per cent of our way and it is certainly much better to advance carefully rather than to make mistakes for which we may be sorry.

All that I have said bears on fusion-welded pressure vessels. The question naturally arises as to the use of fusion welding in other fields, such as in building, bridge and ship construction, where it would be impossible to stress relieve most of the joints. Great progress has been made in this type of welding and the record has been exceptionally clean in regard to failure in service. We therefore should not close our minds to providing more liberal rules for the stress relief of unfired pressure vessels if reliable information is secured which will warrant this being done with safety.

Bursting tests of vessels under hydrostatic pressure made in the ordinary way do not give data on which to base a rule for stress relief, as the residual stresses due to the welding have little or no effect on the bursting strength, whereas tests under intermittent pressures are especially valuable for the purpose. Such tests were proposed a long time ago by the American Welding Society and the Boiler Code Committee in working up a series of tests to be made at the Bureau of Standards in Washington. The tests were designated at the time as "breathing tests" in which the vessels were to be subjected to pressures alternating from about zero to 1½ times the allowable working pressure. These suggested tests were not made at the Bureau of Standards but The Babcock & Wilcox Company made tests along these lines under the supervision of Professor H. F. Moore. These tests did much in establishing the reliability of fusion welding and they are now regarded as accelerated service tests. Tests of the sort should be sufficiently inclusive as a few tests under a given set of conditions may not solve the problem.

Local stress relieving such as heating the area around a nozzle may set up objectionably high stresses in the shell of a vessel adjacent to or around the part that has been stress relieved. Where a vessel is stress relieved in two heats on account of it being longer than can be placed in the stress relieving furnace, there also may be objectionable stresses should the temperature gradient be too high in the portion between the heated and cooled part of the shell. This feature in stress relieving a vessel in two heats was provided for in the preparation of the Joint A.P.I.-A.S.M.E. Code for Unfired Pressure Vessels for Petroleum Liquids and Gases, which includes the requirement that the temperature gradient for the part in question shall not exceed 200 degrees F. per foot along the shell.

There are so many features of stress relief that have not been settled that some users of pressure vessels require all vessels to be stress relieved when the wall thickness is, say, ¾ inch, or over, even though no stress relief is demanded by the A.S.M.E. Code.

My own Company has conducted researches at great expense to investigate the strains produced in fusion welding, which will be published on their completion. These have convinced us that to secure maximum reliability and safety all vessels should be stress relieved after welding. It is our practice usually to stress relieve all vessels even though some are exempted from stress relief by the A.S.M.E. Code.

There is a lot to learn on this subject and while we should be looking forward to the possibility of using more liberal rules, we should make sure not to proceed unless reliable scientific data and facts show that the revised rules will be safe.—"Combustion," May, 1935.

THE TREND OF DESTROYER DESIGN.

By E. Leslie Champness, M.B.E., M.Sc.

(North-East Coast Institution of Engineers and Shipbuilders; November, 1934.)

Whatever class of duty may fall upon the destroyer in wartime, the highest possible speed remains the predominant characteristic of the type, both in offense and to a large extent in defense.

In Figure 1 the principal characteristics of power, cost, displacement and armament on base of length, for destroyer types with 40-knot light trial speed, are shown graphically.

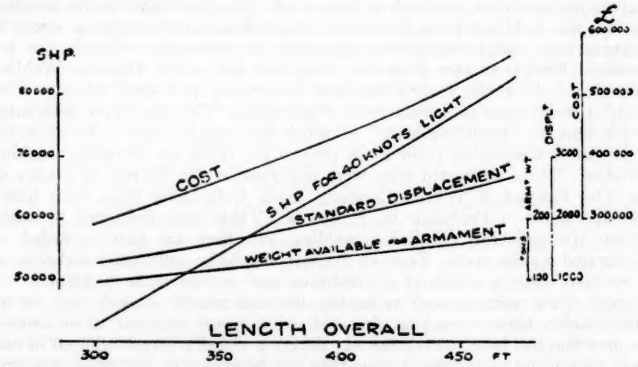


Figure 1.—Approximate Characteristics of 40-knot Turbine Destroyers of Varying Length.

There would seem to be strong grounds for inferring that there is room for two types of destroyer, rather than a compromise all-purpose vessel in which in all probability it may be difficult and certainly uneconomical to carry out all types of duty: (1) The larger scout type, and (2) a development of the smaller type from which designers are departing.

The design of destroyers is perhaps more intimately bound up with questions of both space and weight, concurrently with stability, than that of any other class of vessel. The possibilities of space-saving and the consequent effect on dimensions may be considered under the following headings:—

(*a*) Improved torpedo armament in smaller space;
(*b*) Horizontal separation of superposed guns;
(*c*) Twin gun mountings;
(*d*) Improved boiler developments.

Weight and stability may be affected by the following:

(*e*) Savings from (*a*), (*b*), (*c*) and (*d*) above;
(*f*) Distribution of fuel;
(*g*) Longitudinal framing;
(*h*) Electric welding;
(*j*) Use of light-alloy superstructures.

TABLE I.—PARTICULARS OF 300-FT. (O.A.) DESIGNS.

Design ...	A	B	C	D
Figure ...	2	3	3	4
	2 Boilers	2 Boilers	3 Boilers	Motor
Armament. Gun	← all 4—4·7″ →			
21″ Torpedoes	8	10	8	10
Weights.				
Hull, equipment and armament	662	670	662	670
Machinery	470	470	520	600
Standard displacement	1,132	1,140	1,182	1,270
Oil fuel	350	350	350	150
Reserve feed water ...	18	18	18	—
Deep-load displacement	1,500	1,508	1,550	1,420 tons.
Corresponding draught moulded	10′ 6″	10′ 6½″	10′ 9″	10′ 1½″
G.M. in deep-load condition	1·30	1·30	1·50	2·50ft.
Minimum freeboard amidships	8′ 0″	7′ 11½″	7′ 9″	8′ 4½″
Compare freeboard amidships of war-time destroyers 6′ 3″, and 8′ 6″ in modern larger types.				
S.H.P. Normal	36,000	36,000	36,000	32,000
Overload for 1 hour	41,000	41,000	41,000	37,240
Speeds. Maximum on light trial with oil fuel for 6 hours at start, otherwise complete, at a displacement of	1,185		1,235	1,285 tons
(*i*) with normal power	38½		38	35¾ knots
(*ii*) with overload power	40½		40	38¼ knots
Special light trial without stores or ammunition and with minimum fuel, etc., with overload power at a displacement of	1,080		1,140	1,209 tons
	41½		41	39 knots
In deep-load condition with all oil and with normal power.....	35½		35	34¼ knots

All maximum speeds in smooth water under measured-mile conditions, with no margin.

The results of these considerations are incorporated in designs A to D shown in Figures 2, 3 and 4, and the leading technical particulars are given in Table I, the latter excluding any reductions in weight which would result from items (*g*), (*h*) and (*j*) above.

The increase in deck torpedo armament of destroyers of all nationalities which took place towards the end and after the late War has invariably taken the form of increasing the number of tubes as the number of torpedoes has been increased.

If the fundamental idea be changed so that the increase in number of torpedoes is not accompanied by an increased number of tubes but by a positive and simple rapid reloading device, considerable saving in weight and space can be made with no appreciable sacrifice of military requirements. Two forms of this are given in Designs A and B (Figures 2 and 3), for 8 or 10 torpedoes.

The arrangement of superposed guns in British destroyers was developed from similar arrangements in light-cruiser practice and was soon followed by most foreign navies. There would appear to be possibilities in reducing the horizontal and vertical separation of these superposed guns as now fitted, in the manner indicated for the after guns in Design A (Figure 2).

Protection from attack from aircraft and offense against them are, owing to their speed and the speed of destroyers, likely to be most effective at close range with small-caliber guns with rapid rate of fire, *e.g.*, pompoms, multi-barrelled machine guns, etc. For heavier offense there would seem to be no particular scope for a gun (hand-fed) of caliber intermediate between the pompom and the vessel's main armament. The additional weight and space for such gun, ammunition, etc., seem better allocated to the provision of one or more high-angle mountings of the main gun armament.

While the weight of two guns in a twin mounting is approximately the same as that of two independently mounted guns, the advantage of equivalent armament in reduced space is evident, and the use of such mountings in the compact designs of some recent foreign destroyers is notable. The amount of weight available does not therefore permit of duplication of all guns as at present mounted in the smaller types, while the mounting of four guns in two positions only has some disadvantages in lessening the chance of survival of positions disabled. The compromise which appears most suitable for four guns is therefore:

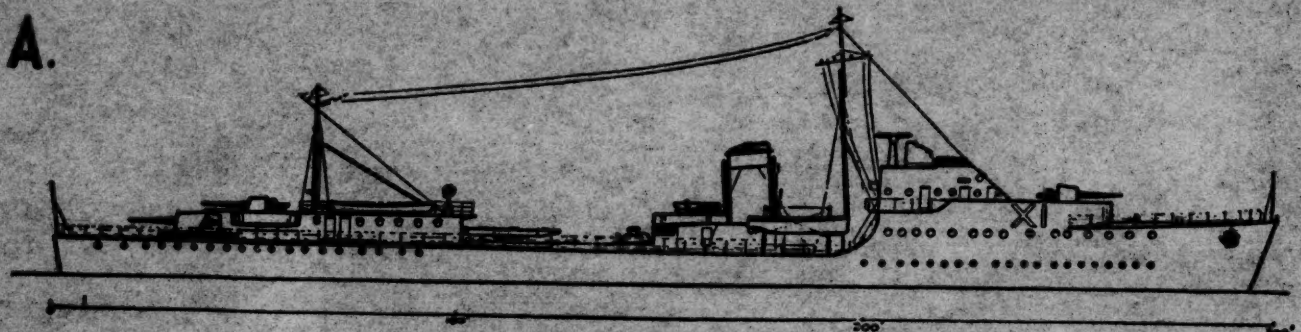
Forward.—Two in a twin mounting, probably with normal elevation mounting, at such a height as shown in the designs, where the top of the bulwark shown is at least equal to the deck level of the upper superposed gun in present modern destroyers. The lower of two superposed guns forward at the forecastle level is frequently liable to be untenable in bad weather.

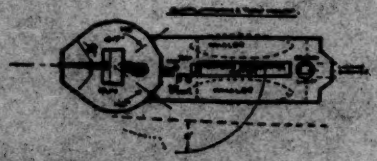
Aft.—Two superposed single-mounting guns, the uppermost being most suitable by reason of its unrestricted arc for a high-angle mounting for heavy anti-aircraft work. The reduced vertical separation is particularly suitable for the provision of the gun-pit necessary for high-angle fire (see Design D after gun, Figure 4).

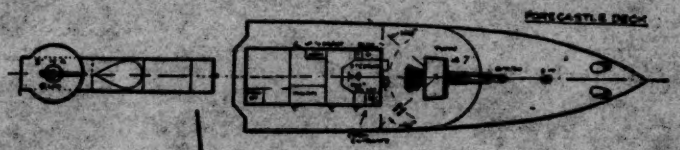
It is hardly visionary to anticipate improvements in boiler equipment enabling powers of 40,000 S.H.P. to be served by two boiler units as in Designs A and B in place of three as in common use, and a comparison between Designs A and C indicates the greater space available for accommodation, etc. The weight saved results in ½ knot increase in maximum speed.

The bridge arrangement incorporated in the designs is proposed for minimum wind resistance, and particular attention is drawn to the arrangement to enable the helmsman to obtain a clear view forward over the No. 1 gun-shield, without either the necessity of raising the whole upper bridge or the use of periscopic arrangements.

A.







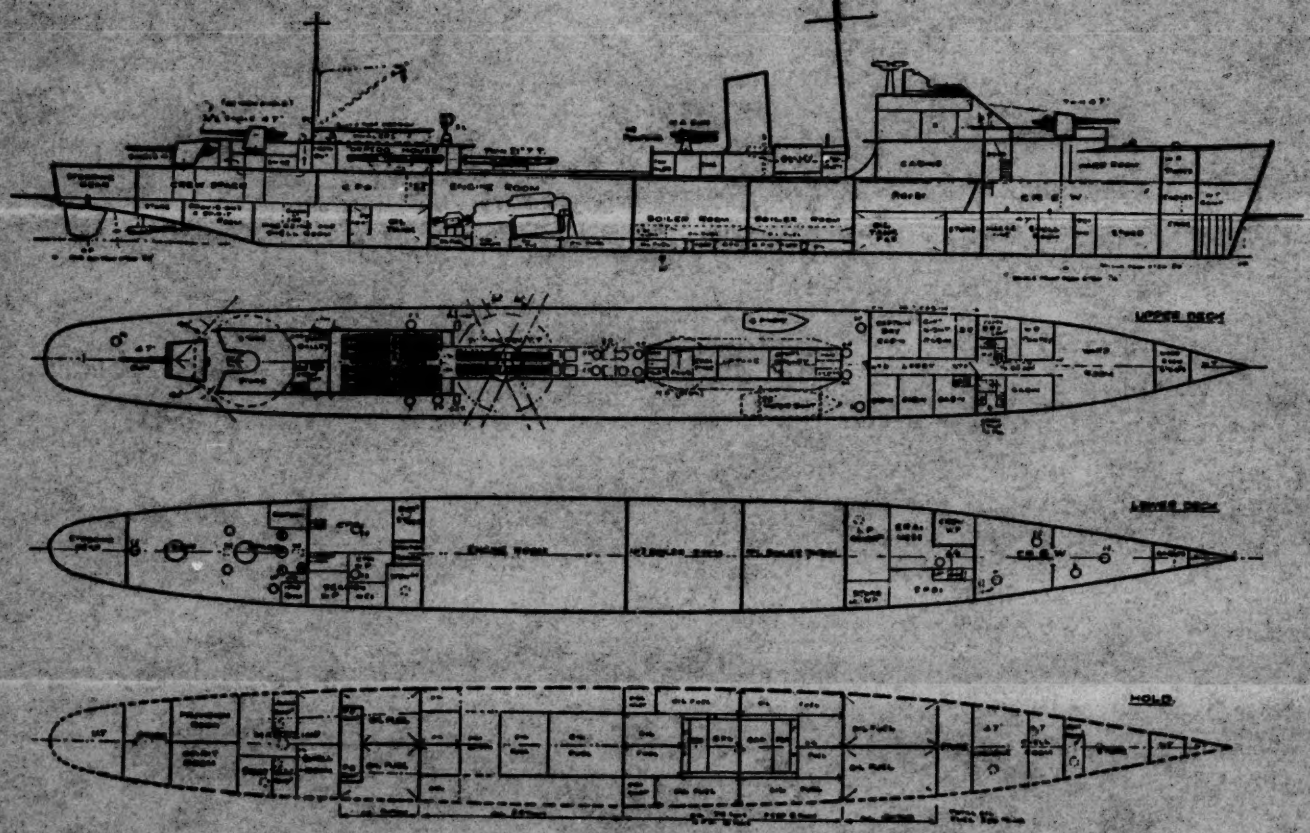
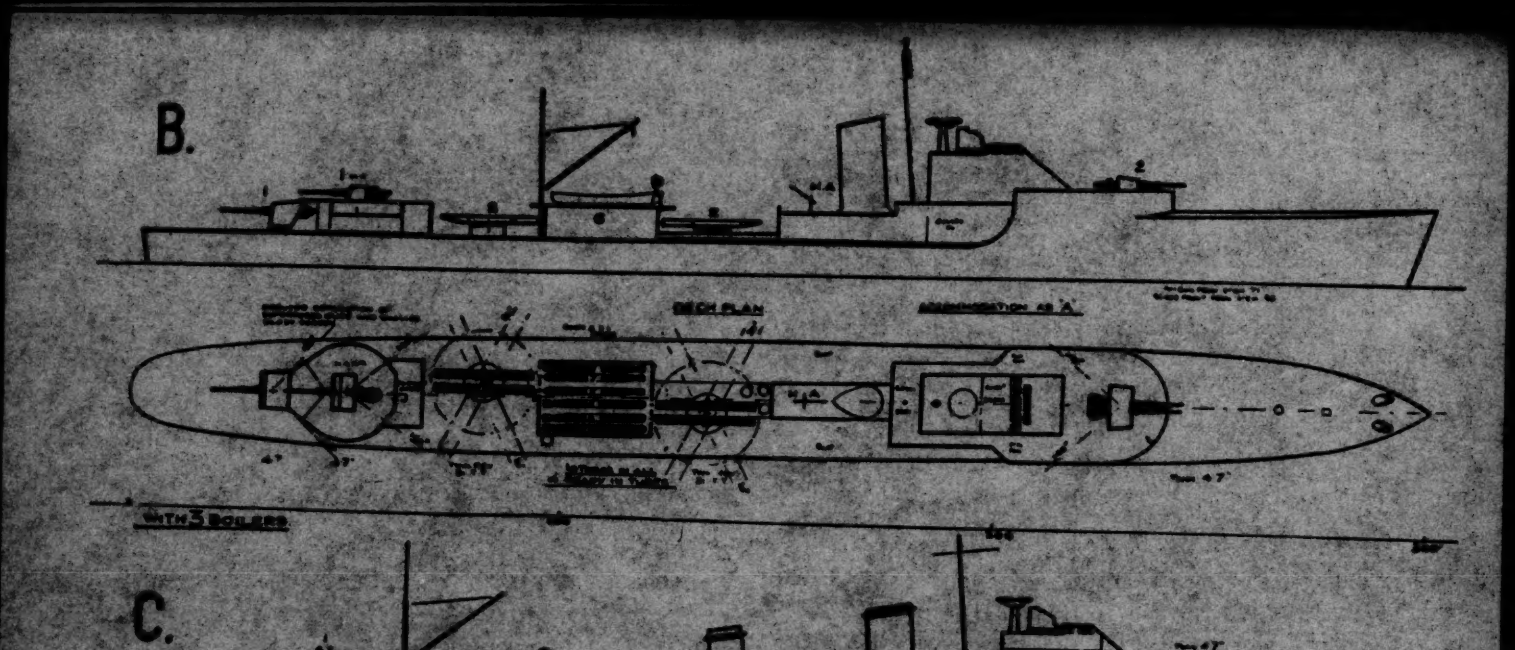


FIGURE 2.—TORPEDO-BOAT DESTROYER—WITH TWO BOILERS; OFFICERS' ACCOMMODATION FORWARD; TORPEDO-LOADING GEAR.

Dimensions: 283 feet B.P and 300 feet O.A., by 30 feet, by 18 feet 6 inches u.d. at side.


B.
DECK PLAN
WITH 3 BOILERS
C.

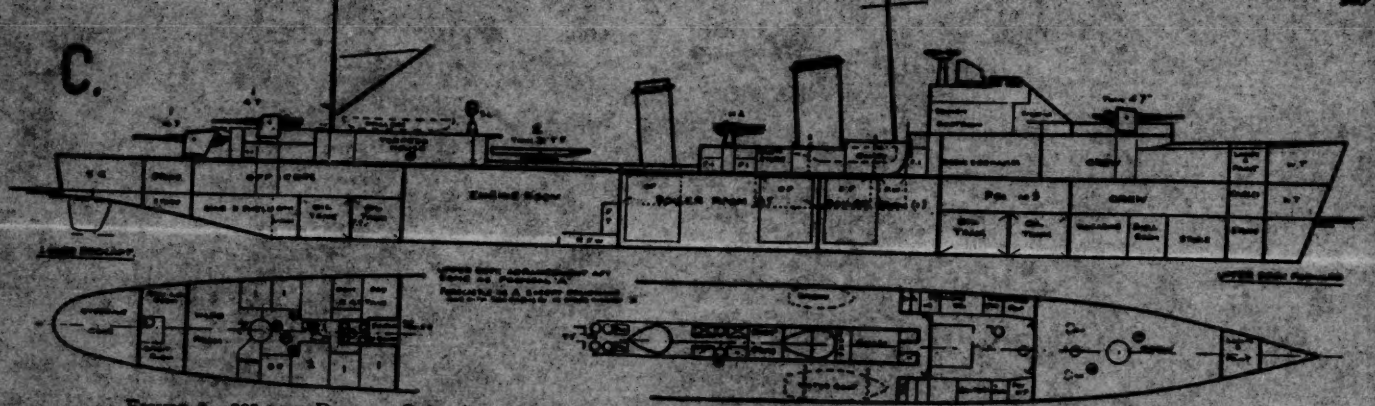
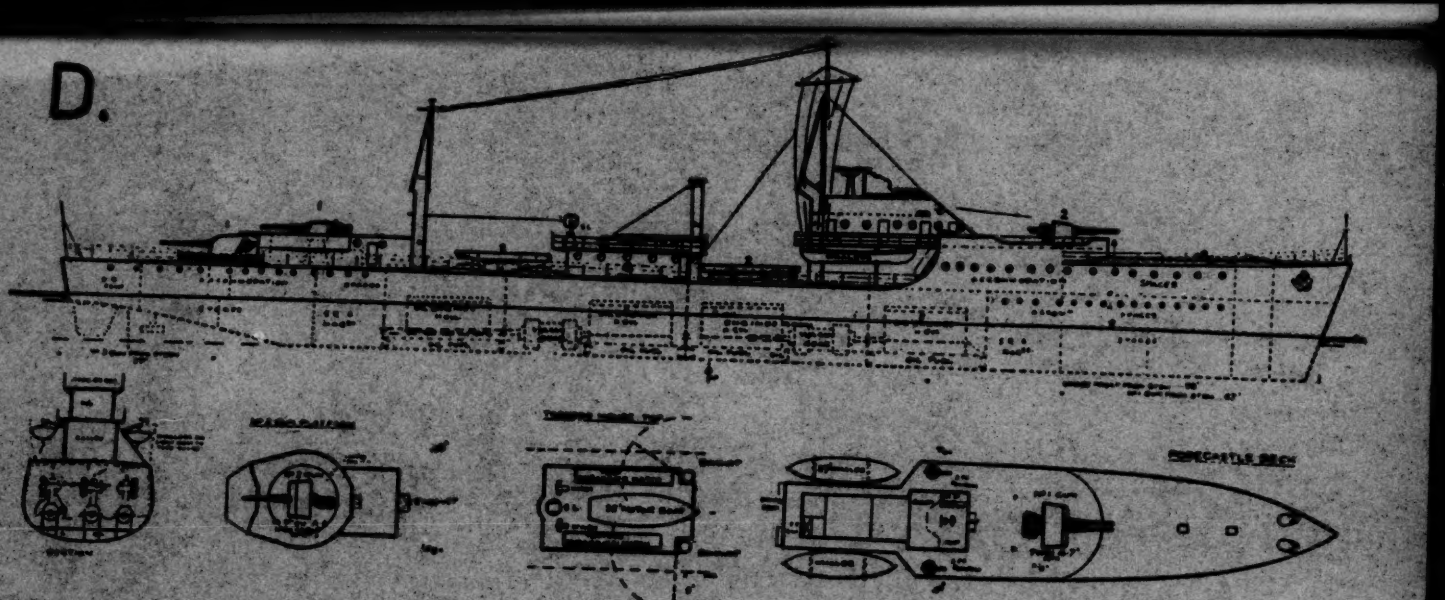


FIGURE 2.—900-FOOT DESIGN O.A. ALTERNATIVE WITH INCREASED TORPEDO ARMAMENT B. AND WITH THREE BOILERS C.

D.



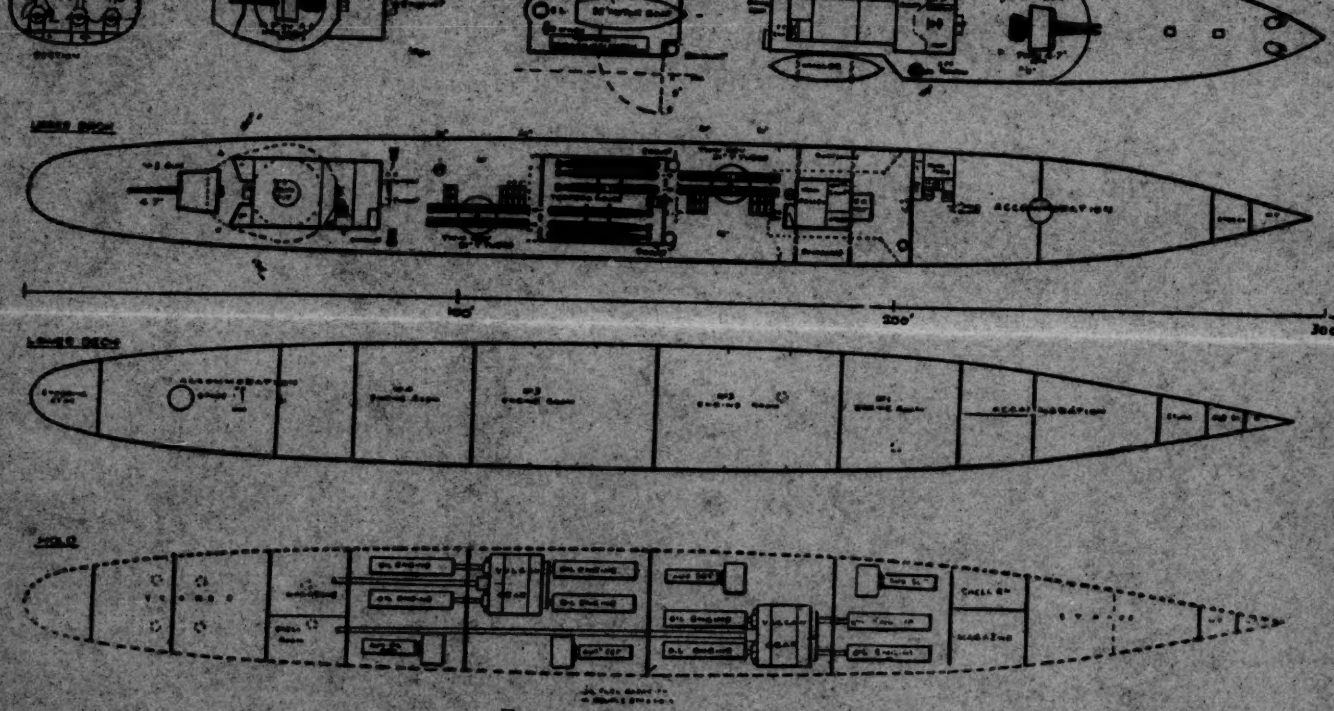


FIGURE 4.—MOTOR DESTROYER.

282 feet B.P. and 300 feet O.A., by 30 feet, by 18 feet 6 inches u.d. at side.

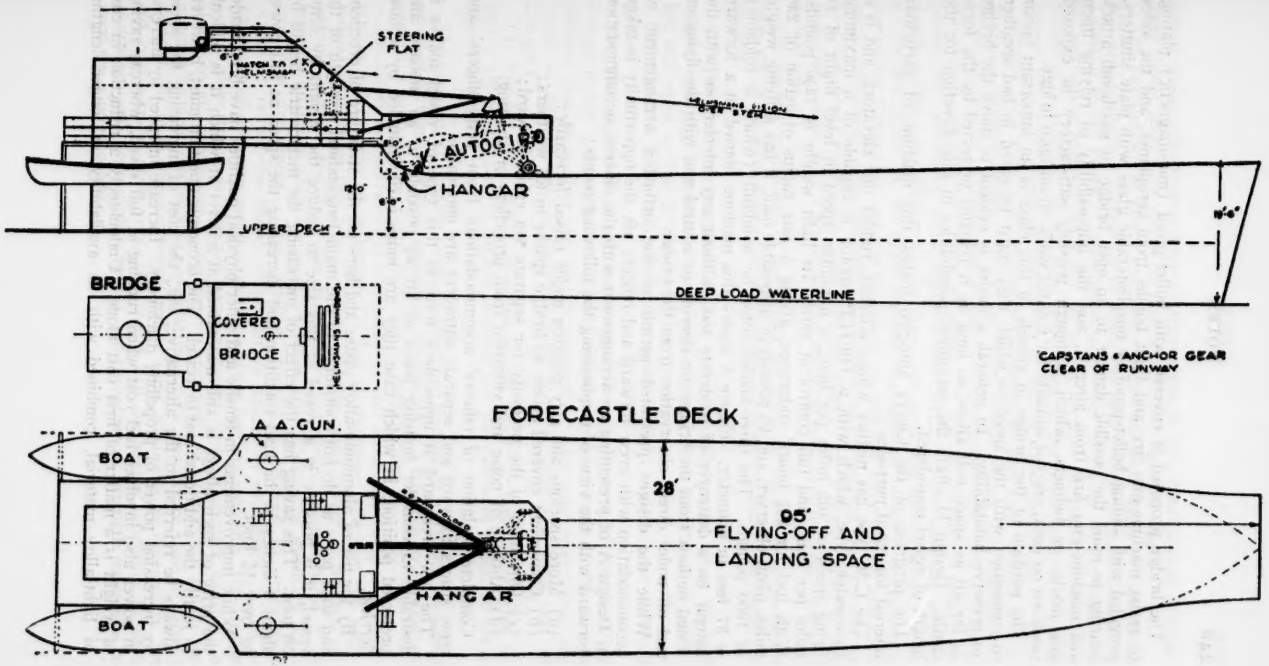


FIGURE 5.—BRIDGE AND AUTOGIRO ARRANGEMENT.

The bridge proposed is covered with bullet-proof (non-magnetic) plating to resist machine-gun fire and light bombs from aeroplanes, and the sides provided with sliding bullet-proof or unsplinterable glass with plate shutters, bearing in mind the possible danger to an open bridge of gas-bomb attack and machine-gun fire from aircraft, and the impossibility of relying upon gas masks or helmets, which, although possibly satisfactory in exposed positions on deck, are not suitable for control work on an open bridge.

The position of the bridge in vessels of this class is an important factor in connection with the speed to which they must be eased in bad weather to preserve habitability. In general, a desire is evident to move the bridge as far aft as space will allow, so long as it remains protected by the forecastle. Design D offers the maximum possibilities in this direction for the length of vessel considered.

The progress of the Cierva autogiro makes this machine of particular interest for naval purposes.

The C.30 type has neither wings, ailerons, rudder nor elevators, and is a two-seater type, which with a 140-H.P. motor is capable of a maximum flying speed of 105 miles per hour, a minimum speed in level flight of 15 miles per hour, and full control at zero. Its light weight is 1250 pounds. With fuel for 2¾ hours' endurance, giving a safe radius of action of 230 miles, pilot, observer, and 95 pounds of disposable load, it has a flying weight of 1900 pounds. The three-bladed rotor or windmill (which is foldable) is 37 feet in diameter. Figure 5 shows this machine housed in a forward hangar on a destroyer of moderate size, without any interference with the visual outlook from the bridge for steering or control and with the flying on and off under direct observation from the bridge.

While the designs presented permit of the orthodox arrangement of accommodation with crew forward and officers aft, the opportunity is taken in Design A of presenting an arrangement with the officers' accommodation forward with the view of emphasizing the following points:

(*a*) More spacious and airy position under raised forecastle;
(*b*) Convenient covered access to bridge spaces in all weathers;
(*c*) Obviation of the necessity for separate sea cabin forward;
(*d*) Absence of noise and vibration from propellers or shafting.

Complete isolation of officers' accommodations from petty-officers' and crew spaces forward and separate entrances are obtainable.

The low freeboard at upper-deck level in this type of vessel makes it desirable to consider housing boats as far as possible in an inboard and protected position, in which case they are more easily handled by a boat derrick.

By the use of aluminum-alloy plates and bars for the structure of erections and such parts as do not contribute to the main longitudinal strength of the vessel, it is possible in a vessel of this size to reduce the weight by some 50 tons. This saving has the effect of increasing the metacentric height by about 9 inches in the load condition and increasing the speed at full power by over ½ knot.

While improvements in tensile and other physical qualities have been made, the price of such alloys is still maintained at a level at which it is difficult to justify the additional cost of vessel. The economic question must, however, always be referred to the alternative, *viz.*, the cost of increasing the speed by increasing power of propelling machinery. Increase in speed by increase in power also involves fuel cost when running at full speed, whereas saving of weight is a matter of first cost alone. Undoubtedly a reduction in cost of light-alloy material, combined with its availability in longer lengths,

would enable constructors to avail themselves of this metallurgical development in the search for means to save weight and increase speed.

The introduction of motor machinery in the German "pocket" battleships and light cruisers, a description of which has been published by Ministerialrat Laudahn in the Press, in association with other means of weight saving, is an interesting example in point. It is not without interest to present the design of a destroyer of the same dimensions as the other designs discussed herein, but propelled by an installation of geared motor machinery.

Design D, Figure 4, incorporates machinery of this type. Eight sets of M.A.N. eleven-cylinder, two-stroke cycle, double-acting oil engines are proposed, capable of developing 33,500 B.H.P. on continuous rating at about 570 R.P.M., driving two shafts at about 350 R.P.M. through Vulcan couplings, and developing continuously 32,000 S.H.P. Four auxiliary engines (each having six cylinders of the same size as in the main engines) drive all the auxiliaries for the main engines—scavenge-air blowers, compressors, pumps for lubricating and piston-cooling oil, fire and bilge pumps, and their own auxiliaries. This machinery is capable of developing (for periods up to three hours continuous work) 39,000 B.H.P. or 37,000 S.H.P. at the propeller shafts, on a running weight complete with systems full, estimated at 600 tons.

The leading particulars of the design are given in Table I, and, compared with the usual three-boiler turbine design, the following main points may be noted:

(1) The machinery space for the motor installation is about 15 feet longer, resulting in smaller accommodation spaces than Designs A, B and C, but still equal to that in many existing destroyers.

(2) About 2 knots less maximum speed in light condition at overload power.

(3) The reduction of maximum speed from light to load conditions at normal power is only about 1¼ knots compared with 3 knots in the turbine designs, due to smaller fuel weight necessary. The mean speed over the range of fuel loading is therefore only about 1 knot less than that of Design C.

(4) For the same radius of action at full speed the total quantity of oil fuel is about half that of turbine designs, and the radius of action at lower speeds is greater.

The economy of running owing to the lower consumption, (about .44 pound per S.H.P. hour including auxiliaries at full power) requires to be weighed against a probable increased first cost of construction.

The closer concentration of machinery or fuel amidships (and/or a reduction in fuel capacity as in Design D) has an important effect on the longitudinal strength of this type of vessel. In the usual design of modern oil-burning turbine destroyer, the highest bending moment experienced is of hogging direction when the main oil tanks are full and the auxiliary tanks amidships are empty. This bending moment is reduced as opportunity occurs to reduce the length of machinery spaces as in Design A, but is accompanied by some increase in the sagging bending moments. In Design D the same result is obtained; and although there is greater equality in stress in all conditions of loading between full oil and empty tanks, the sagging bending moments are of relatively more importance than in present designs. It is in effect a return to the same conditions as obtained in the old coal-burning destroyers, and would call for special attention to deck scantlings and deck girders and a redistribution of strength materials to meet the altered conditions.

With the exception of the *Ardent* built by Messrs. Denny under the 1912-13 Programme, all British destroyers have been framed transversely. No information has been made public giving any comparison of this isolated instance of longitudinal framing with her contemporary sisters framed transversely. There has, however, been much progress since 1912; and although the ends of destroyers are considerably divided transversely, the use of longitudinal framing in the machinery spaces, for the forecastle and other decks, is an economy in weight worthy of consideration.

Similarly, in due course with usual development we may look forward to a saving of weight up to at least 10 per cent of the structural steel weight in destroyer types by the adoption of electric welding generally.—" The Shipbuilder and Marine Engine-Builder," April, 1935.

NEW NAVAL SHIPS.

On Saturday, May 18th, the destroyer H.M.S. *Fury*, the sixth of the eight destroyers of the *Fearless* class, completed her acceptance trials. She was built and engined by J. Samuel White and Co., Ltd., at Cowes. Like her predecessors she is a ship of 1375 tons displacement, and is armed with four 4.7-inch guns, seven smaller guns, and eight torpedo tubes. Her single-reduction geared turbine machinery, taking steam from water-tube boilers, has a trial output of 36,000 S.H.P., corresponding to a speed of 35½ knots. During Saturday's trials very satisfactory results were obtained and the contract speed was exceeded. It may be here recalled that the published trial speeds of British destroyers are the mean speeds of several runs with the vessel complete and in service condition, and without any overload on the machinery. The speeds attained can therefore be reproduced afterwards on actual service and over extended periods, and, unlike some foreign trial results, they represent accurately the sea-going performances of the vessels concerned. Another naval ship, recently completed for the Royal Indian Navy, is the new sloop *Indus*, constructed by R. and W. Hawthorn, Leslie and Co., Ltd. She has a tonnage of 1230, and her propelling machinery consists of two independent sets of single-reduction geared turbines of the latest Parsons type, driving twin screws at about 275 R.P.M. An astern turbine is incorporated in each low-pressure turbine, and cruising stages are provided in the high-pressure turbines to insure a high degree of efficiency over a wide range of cruising power. The machinery is designed to develop a total of 2000 S.H.P. and steam at 250 pounds pressure is supplied by two oil-fired water-tube boilers.—" The Engineer," 24 May 1935.

INFLUENCE OF PROTECTIVE LAYERS ON THE LIFE OF METALS.*

By FRANK N. SPELLER.

National Tube Company (Subsidiary U. S. Steel Corp.).

It is well known that most metals suffer more or less deterioration through reaction with their environment. This may result in surface corrosion or internal disintegration of the metal. Examples of the latter are dezincifica-

* Contributed by the Iron and Steel Division for presentation at the Semi-Annual Meeting, Cincinnati, Ohio, June 17 to 21, 1935, of " The American Society of Mechanical Engineers."

tion of brass and the so-called graphitic corrosion of cast iron. It is also well known that the tendency to corrode varies greatly between different metals and with the same metal in different environments.

Where water is present the mechanism of corrosion is electrochemical and is controlled by many factors, some of which affect the initial reaction of solution of the metal or its tendency to corrode, and others the rate of deterioration over a long period of time. The latter group in this sense have been termed secondary factors to distinguish them from the primary factors that have to do only with the initial tendency. The secondary factors are often very important as they usually determine the ultimate useful life of the metal.

The many factors involved, the various types of corrosion, and means for prevention have been discussed elsewhere and, therefore, will not be referred to in detail here.[1]

CLASSIFICATION OF CORROSION TYPES.

The corrosion problem is now recognized not as a single one, but rather as a group of problems complicated by many variables, the resultant of which determines the rate of attack. In view of the fact that factors external to the metal usually predominate, it was found convenient to classify corrosion according to the environment and we now speak of it as atmospheric, underwater, soil, or chemical corrosion, or electrolysis due solely to stray electric currents. While there are many factors involved in each type of corrosion, usually one or another exercises the controlling influence. For instance, in atmospheric corrosion the predominant factor is moisture; in water, the oxygen concentration; in soil, the electrical conductivity,total acidity, drainage, etc. Some special problems apparently do not fit into this classification. In some petroleum products water is but slightly soluble, yet the free oxygen may amount to 30 parts per million or more (about five times the amount present in sea water under normal conditions). Therefore, where gasoline containing oxygen is stored in steel tanks and pipe lines and water is present on the surface of the metal, the conditions are highly favorable to corrosion, since the oxygen in the gasoline is transferred to the water. In this case (as in atmospheric corrosion) an excess of oxygen is available. Therefore, in gasoline pipe lines the controlling factor is the amount of moisture in the gasoline, which is usually quite limited. Probably the most economical remedy in such cases is to dehydrate the gasoline.

PRINCIPAL FACTORS AFFECTING RATE OF CORROSION IN WATER.

The principal factors affecting the rate of corrosion in the presence of water are listed in Table 1. They have been grouped in this manner to indicate conveniently whether they are associated mainly with the metal or the environment.

[1] "Corrosion—Causes and Prevention," by F. N. Speller. Second edition. McGraw-Hill Book Co., 1935.

TABLE 1. PRINCIPAL FACTORS AFFECTING CORROSION IN PRESENCE OF WATER.[2]

Factors associated mainly with the metal	*Factors which vary mainly with the environment*
Effective electrode potential of a metal in a solution	Hydrogen-ion concentration (pH) in the solution
Overvoltage of hydrogen on the metal	Influence of oxygen in solution adjacent to the metal
Chemical and physical homogeneity of the metal surface	Specific nature and concentration of other ions in solution
Inherent ability to form an insoluble protective film	Rate of flow of the solution in contact with the metal
	Ability of environment to form a protective deposit on the metal
	Temperature
	Cyclic stress (corrosion fatigue)
	Contact between dissimilar metals or other materials as affecting localized corrosion

Some of these factors, such as the standard single-electrode potential and homogeneity, affect the *initial* rate, while others, such as temperature, velocity of flow, and available oxygen, more often determine the *average* rate at which the corrosion process proceeds; and still another group, which includes mainly concentration cells and contact effects, determines the *distribution and localization of corrosion*. The latter frequently results in penetration of the article before the metal has lost 5 or 10 per cent of its weight.

METAL SURFACE FILMS.

Space does not permit a discussion of all the dominant factors, but as films are evidently of fundamental importance to the life of metals they will be referred to in more detail.

The environment, as a rule, influences both the actual and relative rate of corrosion of a metal much more than the ordinary variations found in the metal. It has been demonstrated that all base metals such as iron, copper, aluminum, etc., acquire a surface film when exposed to air at normal or elevated temperatures, and that it is usually the resistance of this film that determines the life of the metal rather than some internal property, inasmuch as the metal itself usually does not come into direct contact with the attacking medium. Protective surface films are formed by reaction between the metal and its environment, usually by oxidation. The materials essential for film-building may originate mainly in the metal or in the environment, as in passivation with nitric acid or chromates; or both metal and environment may contribute their share in building the film, as in stainless steels.

Metals, such as chromium, nickel, copper, and aluminum form highly resistant films in some environments but not in others, and when alloyed with iron in solid solution have the power of conferring on it some of their film-

[2] Factors that control the rate of corrosion may also be classified according to their influence on the anodic or cathodic reaction. For instance, the former is stimulated when the corrosion products are dissolved, peptized, or are permeable to ions; while the latter is stimulated by free oxygen (the most common cathodic depolarizer), or by materials in contact with the metal that lower the overvoltage and facilitate the liberation of free hydrogen. A discussion of the mechanism of corrosion along these lines will be found in a paper by Evans and Hoar, "The Mechanism of Metallic Corrosion," Trans. Faraday Society, vol. 30, no. 156, part 5, May, 1934.

building power according to the amount of alloying elements present. Chromium is the most useful of these metals, and, when alloyed in the proper amount and correctly treated, ferrous alloys are obtained having superior physical properties combined with high resistance to corrosion. The determination of the amount of film-forming constituents that will give the maximum corrosion resistance to iron at the lowest cost is a promising but difficult field of research.

In referring to unsolved corrosion problems last year[3] the author referred to the problem of metal surface film as follows:

The study of the formation, structure, and protective properties of metal surface films and the nature of their bond to the metal is one phase of this problem, the solution of which may be of far-reaching importance, for corrosion is usually a problem in surface reactions. To afford permanent protection it is, of course, essential that films be self-healing when injured. This is another property that requires more investigation. Further study of protective metal film may also help to produce better artificial coatings. For instance, the failure of paints and certain metallic coatings usually starts at pinholes. In certain environments small imperfections in coatings have been found clogged with corrosion products, and in such cases the coating has shown a much longer life. It has been suggested that the bare metal and coating material may perhaps be designed so that the initial corrosion products will seal discontinuities in the coating.[4] This suggests another promising field for research.

In discussing the reltatively simple problem of oxidation of steel at elevated temperatures, Dr. Johnson has raised the following questions which apply equally well to the whole problem of film protection.[5]

The information now available furnishes little indication of what would be a logical procedure in a search for a ferrous alloy highly resistant to scaling at elevated temperatures. The obvious requirements are that the alloy should form a closely adherent surface film of oxide, and that this film should be nearly impervious to the passage of oxygen atoms inward or of metal atoms outward; but as to the implications of these requirements, one can raise questions which can be answered only by specific investigation. For instance, does close adherence of scale imply that the pattern of the metal atoms in the scale film is nearly identical with the arrangement of those metal atoms in the surface skin of the specimen? Or even that there is one network of metal atoms common to both scale and metal surfaces at the interface? Or does it imply merely that the effective expansion coefficients of scale and metal in the plane of the interface must be nearly alike? Should the alloying element have a high rate of diffusion through the iron matrix, so that it comes rapidly to the surface to heal up imperfections in the scale? And must it be more readily oxidizible than iron, as are chromium and aluminum? Is a highly impervious oxide layer one in which oxygen or metal is almost insoluble? Is it an oxide of a metal which forms a single stable oxide, as aluminum oxide, rather than of a metal such as iron with its several oxides and oxide mixtures? And, is it likely to be a compound oxide, such as a spinel? How far is the success of the protective skin determined by the character of the very first element of the film formed on the metal surface? On questions such as these, knowledge is at present largely or entirely lacking; until some of them are answered, one is limited to purely empirical tests, most of which are certain to be unsuccessful.

[3] "The Corrosion Problem," by F. N. Speller, Howe Memorial Lecture, A.I.M.E., February, 1934.

[4] Op. cit., p. 1.

[5] "Corrosion Problems," by John Johnston, "Industrial and Engineering Chemistry," Dec., 1934, p. 1240.

The answer to these, and similar questions, must await further investigation of the how, why, and wherefore of films on metallic surfaces. Such investigations would throw light not only upon how to develop the best protective film on each metal, but also upon what really constitutes a good adherent coating, whether of paint or metal, put on by dipping or by electroplating, or of enamel.

The influence of various kinds of metallic crystalline architecture on the bond and stability of metal surfaces might be another interesting field for investigation.

The stability of films has been found by Vernon[6] and Evans[7] to vary materially with the conditions of initial exposure. When the environment is very favorable to the formation of a dense adherent film, unusual resistance to corrosion has been obtained even with ordinary wrought iron and steel, which admittedly are vulnerable under usual conditions. This fact seems to explain the long life occasionally found with old wrought iron and steel.[8]

Several cases have been reported where, on removal of the initial corrosion products from well-preserved iron and steel and exposure of the clean surface to ordinary air, rust developed rapidly. The nature and stability of metal surface films are now considered to be among the most important factors determining the course of corrosion. Local differences of potential due to concentration cells or contact with dissimilar materials may be sufficient to break down the film and cause pitting, even on stainless steel. Therefore, it is essential in running extensive corrosion tests to draw time-rate curves so that the trend can be studied and predicted beyond the limits of the test period. Frequently, the rate of attack decreases and sometimes corrosion ceases altogether after a certain time, due to the formation of a protective layer of inert materials on the metal (Figure 1). If the latter did not take place, the life of most metals would be much shorter.

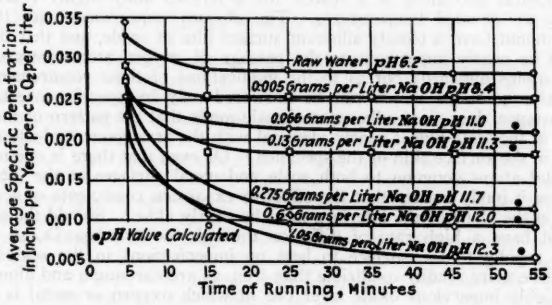


FIGURE 1.

[6] W. H. J. Vernon, (*a*) "Investigation of the Atmospheric Corrosion of Non-Ferrous Metals, Second (Experimental) Report to the Atmospheric Corrosion Research Committee, British Non-Ferrous Metals Research Association," Trans. Faraday Society, vol. 23, 1935, pp. 113-185; (*b*) "The Role of the Corrosion Product in the Atmospheric Corrosion of Iron," Trans. Electrochemical Society, vol. 64, 1933, p. 31; (*c*) "A Laboratory Study of the Atmospheric Corrosion of Metals," Trans. Faraday Society, vol. 27, 1931, pp. 255-277.

[7] U. R. Evans and associates, "The Passivity of Metals" (A series of seven papers), Journal of the Chemical Society, London, 1927-1932.

[8] Archeological relics have been unearthed on the site of Dura-Europos on the Euphrates River, by the Yale-French Academy Expedition which indicate that the making of wrought iron on a commercial scale dates back 1800 years from the present time. ("Iron Age," Jan. 10, 1935, p. 13.) The oldest samples so far discovered originated about 1000 B. C.

MEANS OF PROTECTION IN GENERAL.

Prevention of corrosion has usually been accomplished in general by (1) improving the inherent resistance of the metal; (2) changing the environment; and (3) isolating the metal from destructive environment. In the last analysis all of these depend upon the formation of a resistant protective layer of some sort that will retard or stop the corrosive process. In a few cases corrosion has been arrested by impressing an electric current on the metal, as in the recently developed "cathodic protection" of underground pipe.

The prevention of corrosion of iron and steel in their various commercial forms is often a matter of great importance. The physical properties of these metals can be so well and economically adapted to common requirements that they are well nigh essential to our comfort and civilization. It is, therefore, not surprising that the tonnage in use is nearly twenty times that of all other metals together. Unalloyed iron and steel, unfortunately, have a rather limited ability to protect themselves, except under certain special conditions as in dry air. Nevertheless, these useful metals have in many cases given good service for forty years or more under varying conditions of exposure with little or no artificial protection. Rails and underground pipe lines are typical examples. It is usually desirable, however, to apply paint coatings to the metal when exposed to the atmosphere, if only for the sake of appearance.

When metals are "idle" they often show greater depreciation from corrosion than when in continuous service; therefore, deferred renewals or inadequately protected iron and steel, unused during the years of the recent depression, are likely to be quite a factor in restoring normal production.

Stainless steels (steels high in nickel and chromium such as the well-known 18-8) are the best examples of what may be accomplished by applying the first principle of protection already referred to. To secure the best results, the alloying metal should be retained in solid solution in the iron. These alloys exhibit, in some corrosive environments, marked ability to build up a highly resistant surface layer which quickly arrests corrosion.

The second principle of protection is illustrated by the use of neutralizing reagents, passivators, and various means for removing oxygen from water. These expedients have proved useful and economical in controlling corrosion under various conditions in practice. This type of protection cannot be discussed in detail within the scope of this paper, although it depends in principle upon the formation of a resistant layer (solid or liquid) on the metal.

The third type of protection, artificially applied coatings of resistant materials, is the one most generally employed and is therefore discussed in more detail.

FILMS AND COATINGS.

In discussing metal-surface protection, it is convenient to consider the subject under three headings, a classification which is based on the thickness of the protecting layer:

(1) Thin films which may be invisible and sometimes only a few molecules in thickness.

(2) Metallic and non-metallic coatings, such as are obtained by galvanizing with zinc or by painting.

(3) Coatings of more substantial thickness, such as reinforced hot-bituminous or portland-cement mixtures.

FILM PROTECTION.

Much more is known about the physical properties and the structure of metals than about their surface condition under various exposures, yet it is

evident, at least in the case of the metals in common use, that we are dealing with a film-covered surface and not with a real metallic surface. Without the natural tendency to build up surface films that are often self-healing, all but the rarer noble metals, such as gold or platinum, would have a much more limited life under most service conditions. The growth of these very thin invisible films has been demonstrated mainly by their effect in depressing the potential of the metal, while in other cases their presence has been demonstrated by dissolving away the base metal so that the film is rendered visible under the microscope.

Fortunately the film-building properties of steel can be improved to almost any degree desired by alloying this metal with other metals, such as chromium, nickel, and copper, which impart to iron some of their own inherent characteristics in this respect. Chromium, although more oxidizable than iron, is an ideal metal to increase the corrosion resistance of iron, as it forms a solid solution with iron, and, up to about 20 per cent, the alloy is readily forgeable and has good physical properties when properly treated in fabrication. When 12 per cent or more chromium is present in solid solution in iron (*i.e.*, one atom of chromium to eight atoms of iron) an exceptionally strong and adherent film is developed which results in an abrupt improvement in corrosion resistance in certain environments, such as moist atmospheres.[9]

The addition of as little as 0.15 per cent copper increases the resistance of bessemer steel four or five times in industrial or seashore atmosphere, and by increasing the copper to 0.4 per cent and adding about 1 per cent chromium with 0.8 per cent silicon and about 0.15 per cent phosphorous, the resistance compared with "copper steel" can be more than doubled in the atmosphere and in salt water, and at the same time the tensile strength is increased by 50 per cent. The addition of 3 to 5 per cent chromium provides another series of ferrous alloys at moderate cost that have very desirable physical properties and considerable resistance to corrosive sulphur compounds and oxidation, and therefore are now much used, as for instance, in oil-refinery equipment.

The well-known stainless steels, of which 18-8 chromium-nickel steel is the most widely used, are the most highly developed of the rust-resisting ferrous alloys, but no one metal has been found nor is likely to be found that will serve all purposes economically.[10]

Protective films can also be formed on ordinary iron that has no inherent power to protect itself by the use of inhibitors, such as soluble chromates or alkalies. The inhibiting power of these substances depends on their ability to react with the metal and form a solid protective film or inhibiting layer of solution on the metal surface. This principle has been applied to the prevention of corrosion from brine in refrigeration, and from recirculating cooling water in condensers, by the addition of sodium chromate and by the well-known alkaline reagents in boiler-water treatment, and is often found to be more economical than the substitution of a more expensive metal or the application of a protective coating.

PROTECTIVE COLLOIDS.

Fine particles dispersed in water, such as clay colloids in slightly alkaline solutions, form protective deposits that sometimes greatly retard corrosion. The adsorption of hydroxyl ions on the surface of colloidal particles probably assists by increasing the alkalinity of the solution in contact with the metal.

[9] "The Structures of the High Chromium Stainless Steels and Irons," by E. C. Bain, Year Book of the American Iron and Steel Institute, 1930, p. 271.

[10] Duplex metals have been made for some purposes by welding a layer of 18-8 steel (Plykrome) or pure nickel (Ni-clad) about 1/32 inch thick on a carbon-steel base. Such combinations are readily fabricated into many forms by fusion-welding with the right kind of wire and suitable technique.

For example, steel water mains in the Middle West have been found free from corrosion under a layer of absorbed particles of clay about 1/16 inch thick. The author has also found this type of protection to be highly important in connection with the use of steel drill pipe. The mud-laden fluid, which circulates through the pipe, is usually conditioned with colloidal material where this is not formed in sufficient quantity in the course of drilling. To stabilize the colloids the fluid is maintained at the proper alkalinity, ranging between a pH of from 8 to 10, according to the nature of the fluid. Fortunately, mud conditioning favorable to good drilling practice has been worked out, but apparently without any thought of preventing corrosion. This practice which affords considerable protection against corrosion, therefore, explains why failures from corrosion fatigue are relatively a rare occurrence in rotary drilling.

OXIDE COATINGS.

Oxide coatings, such as the Bower-Barff or the coating formed on iron in a hot phosphate bath by the Parkerizing or Bonderizing processes, or the dense adherent layer of corrosion products that form on aluminum or copper steel in air, like the thinner surface films already discussed, are so firmly attached to the metal as to suggest some kind of atomic bond, the nature of which has not yet been discovered. The adherence of paints and lacquers is materially improved by applying them to surfaces that have been so treated as to form a tight bond between the surface and the coating.

PAINTS AND LACQUERS.

About 80 per cent of all iron and steel products require some kind of protection if only for the sake of appearance. For these metals under atmospheric exposure, about 100,000,000 gallons of paint are used annually on the average. The quality of these paints has been notably improved in recent years. For uses under atmospheric exposure, paints consist essentially of (*a*) a vehicle, usually linseed oil with a little oxidizing material to accelerate drying; and (*b*) various pigments to improve the strength and weather resistance of the coating. Basic pigments, such as the much-used red lead, and lead or zinc chromate, inhibit corrosion when applied next to the metal, and therefore are commonly used as a priming coat. That this useful property actually exists in pigments has been proved by the fact that when paint films containing 20 per cent or more of these compounds were scratched and the metal exposed, it was found that corrosion had been greatly retarded. It is mainly for this reason that red lead has given such good results in primers. The coating next to the metal should be thin enough to wet the entire surface, including all depressions and cracks. The outer coatings should be made less fluid by the addition of a larger proportion of pigment. For use under atmospheric conditions, the outer coating should be designed mainly to assure durability and the exclusion of water. The addition of two pounds of flake aluminum per gallon of vehicle (preferably spar varnish) has given good service as an outer coat when exposed to weather conditions.

Synthetic resins of the phenol-formaldehyde type are extensively used in the manufacture of enamels and clear varnishes. They are quite resistant to the effect of exposure to water and usually give excellent service. Other synthetic resins, made from the reaction of phthalic anhydride and glycerin, such are the alkyd-type resins, are generally plasticized with fatty acids of oils. These resins, when pigmented, find a wide use in the automotive industry. They can be sprayed or brushed, and produce surfaces of high gloss and of excellent weather resistance. However, their resistance to water is not usually as great as properly formulated varnishes of the phenol-formeldehyde type.

Chlorinated rubber and other somewhat analogous rubber compounds have appeared recently in the paint industry. These compounds produce coatings which are quite resistant to dilute acid and alkali solutions. The unsatisfactory adhesion which such coatings sometimes give when applied to ordinary metal surfaces may be overcome through the use of properly formulated tung-oil compounds as plasticizers.

Lacquers are distinguished from paints in that the lacquer vehicle hardens entirely by evaporation of a volatile constituent and not by oxidation.

In the nitrocellulose-lacquer industry, many advances have been made during the last few years. Hundreds of new plasticizers and solvents have been utilized, and the formulations are extremely varied. As a rule, low-viscosity nitrocellulose is plasticized up to one-half its own weight with such organic plasticizers as dimethyl phthalate, dibutyl phthalate, or tricresyl phosphate. Various grades of natural and synthetic resins are employed to give adhesion and gloss. A very wide range of solvents of low, medium, and high boiling points is utilized to secure the desired speed of evaporation.

Assuming that proper care in application has been taken so as to get a uniform adherent film of the right thickness, the life of painted metal depends upon (*a*) the nature of the metal; (*b*) freedom from foreign matter between the metal and paint, such as hydrated oxides of iron (rust), moisture, and soluble salts; (*c*) the proper selection of suitable vehicles, pigments, thinners, and driers properly blended; and (*d*) the atmosphere. The formulation of paints and lacquers calls for a high degree of technical skill which is readily available in the paint industry. As paint films are slightly pervious to moisture, anything that interferes with perfect adhesion to the metal or that promotes corrosion under the paint will bring about early failure. Hence it is most essential to secure a clean dry surface. The first coat should be applied immediately after the metal is cleaned. The initial film formed in the air may be an oxide which will be subsequently hydrated and thus loosen the bond. In some cases it has been found of advantage to sweep the metal with the flame of a gasoline torch to thoroughly dry the surface just before painting. Such coatings when applied on warm surfaces of newly rolled steel have proved to be highly durable. An outer water-proof coating over an inhibitive base coat is usually quite effective, and two or three coats are naturally much better in this respect than one.

For under-water service the precautions referred to should receive special consideration. Wherever practicable, the paint should be thoroughly dried before immersion, as thereafter there is no chance for further drying by air. It has been found that more artificial drier can be safely used in paints for under-water service than under atmospheric exposure. When the drying time is limited for economic reasons, as, for example, when painting ship hulls in dry dock, quick-drying paints of high durability should be employed.

For the painting of steel hulls for fresh-water service, the following formula has proved satisfactory: Composition of pigment: red lead 75 per cent; lead chromate 12.5 per cent; zinc chromate 12.5 per cent. Composition of vehicle: treated china-wood oil, 100 per cent (volatile, 63 per cent, non-volatile, 37 per cent).

This mixture dried to the touch in one hour and the hull was immersed 24 hours after the paint was applied to a freshly sandblasted surface. The sandblast cleaning was done at a cost of two cents per square foot. After two years in service this coating seems to be holding tightly to the metal in pit holes caused by previous corrosion. It is interesting to note that serious pitting occurred on the hulls of ore carriers that were laid up for two years during the depression, but no material corrosion was found on the unpainted hulls of these ships prior to that time when they were in regular operation eight months of the year.

Paints when used under water often fail by blistering. This is probably due to absorption of water and to building up of osmotic pressure under the paint films. Evidence of any tendency to blister can be obtained by immersing test panels for a month or two after they have been cleaned and coated in strict accordance with the procedure that will be followed in practice. A relatively impermeable and strong outer coat of a highly durable paint applied over the wrong kind of primer has blistered seriously under water. Pure red lead and some of the rubber derivatives or phenol-formaldehyde resin paints have given the best results in fresh-water immersion tests.

Much more is known about the formulation of paints for atmospheric service than for under-water service. Two coats of different paints applied with a limited drying time, have a tendency to cause blistering in water, and therefore, single coats have sometimes proved more durable than double coats under these conditions. For submerged service it may prove better to make the priming coat relatively less permeable than the outer coats. Improvements in paints for use under water may be expected as a result of work now in progress.[11]

TESTING OF PAINTS.

While there have been many practical tests of paint coatings under atmospheric conditions, much less has been done on paints for use under water or at the water line. It would seem to be highly desirable to have definite specifications for procedure in testing with respect to the important factors that determine the life of the coating in air, in water, and at the water line. Tests should be made under all these conditions to secure the most complete data on the behavior of paint coatings in each environment.

The testing records should include general information as to all chemical and physical characteristics of the paint and the drying time of each coat. Atmospheric conditions should be recorded and should be normal, especially as to humidity, when the paint is applied. The specification should also include detailed instructions as to the composition and finish of the bare steel, the preparation of a clean surface, and method of application of the paint. Laboratory tests should be made to indicate the adhesive properties, the resistance to impact of each coat, and its permeability to water. The time of drying of each coat should be specified, and, where the paint is to be immersed in water, the time between the application of the last coat and immersion of the finished test panels should be specified so that each paint under test will be treated in the same manner. In testing paints for reconditioning ship bottoms or for use in other locations where the drying time is limited, this limitation, of course, should be followed in preparing test panels.

The corrosion-resisting properties of the base metal have been shown to influence the rate of destruction of paint coatings. For instance, paint applied to copper steel in freight-car construction has been shown to last much longer in service than when applied to ordinary steel, and the serviceability of paint might be expected to be still better when more resistant steels are used. Hence, it would seem important to carry on exposure tests with bare and painted metals in the same location and at the same time for, after all, most steel construction is painted, if only for the sake of appearances, and the life of the painted surface is of more practical interest than that of the bare metal.

Petroleum grease, wax mixtures, or similar non-drying compounds, when well rubbed on a clean surface, have given good results in protecting steel

[11] Information regarding paints for special purposes may be obtained in circulars of Scientific Section of the Institute of Paint and Varnish Research, Washington, D. C. See circular No. 331, June, 1928.

exposed to water, as in water ballast tanks on ships or in damp coal mines where the coating is liable to be injured by abrasion. Elastic water-proof paint films of the drying type have been successfully applied on a slushing grease base coat.

COATINGS FOR PIPE UNDERGROUND.

The principal object of a protective coating on pipe is to reduce maintenance costs as far as possible. This is particularly important when the pipe is laid underground and where the coating cannot be easily renewed. The efficiency of a coating will depend on the life of the coated pipe as compared with bare pipe. This in turn depends upon the thickness and character of the coating and the physical and chemical properties and variation in water content of the soil. The question, "What is the best coating for use underground?" is therefore largely an economic one.

The American Petroleum Institute and the American Gas Association have placed research associates at the Bureau of Standards. This cooperative association has conducted two independent tests. The first, sponsored by the American Petroleum Institute, includes 46 coatings on short sections of pipe and 19 coatings on operating lines. The American Gas Association has sponsored a series of tests including 42 kinds of proprietary coatings applied to 2-foot lengths of 2-inch pipe. These coatings were tested in typical soils, the corrosive character of which had been previously tested by exposure of bare metal for several years.[12]

RESULTS OF BUREAU OF STANDARDS' TESTS.

The facts so far established as a result of the Bureau of Standards' tests on coatings and bare metals are:

(1) The chief causes of coating failures arise from injuries to the coating before the pipe is laid in place, from distortion by soil stress, and from deterioration of the coating material.

(2) Very few of the present commercial coatings are entirely adequate to prevent underground corrosion under all soil conditions.

(3) None of the commercial coatings are proof against electrolysis, although many of them offer additional resistance.

(4) The relative behavior of coatings is different in different types of soils.

(5) All coatings fail to protect in some corrosive soils, but by making a judicious selection corrosion may be greatly reduced. Therefore, it pays in many cases to lay the pipe bare or to apply only a thin and imperfect coating of low cost over the whole line, rather than a more costly one, with the expectation of reconditioning the line after failures occur.

(6) Paints or single-dip bituminous coatings are of very little use for underground protection, except that they may limit the area under attack.

(7) Reinforced or shielded fabrics improve bituminous coatings, but the thickness of the coating should be not less than 0.125 inch in corrosive soils.

(8) Organic fabrics tend to rot in certain types of moist soil containing certain kinds of bacteria. Saturated asbestos felt has proved satisfactory under these conditions.

(9) Coal-tar pitch has sometimes been found more durable and less permeable to water than petroleum asphalt.

(10) Hot-galvanized zinc coatings afford considerable protection in some soils, their value depending upon the thickness of the zinc and the character of the soil.

[12] Bureau of Standards Research Papers Nos. 95, 329, 359, and 638; and Technical Paper No. 368.

(11) Cathodic protection of a pipe line that has been given a bituminous coating of high electrical resistance has afforded complete protection at moderate cost in limited soil areas having relatively high conductivity.

(12) With respect to the bare metal, soils vary considerably in their corrosivity. In many soils the action between the metal and the soil is not severe. No material difference in corrosion penetration is found between the various commonly used types of ferrous metals in any of the corrosive soils, although in the more corrosive, the rate of penetration varies greatly with the character of the soil. As indicated in Figure 2, the rate of penetration of the metal in the many types of soil usually decreases with time, rapidly at first and then more slowly.

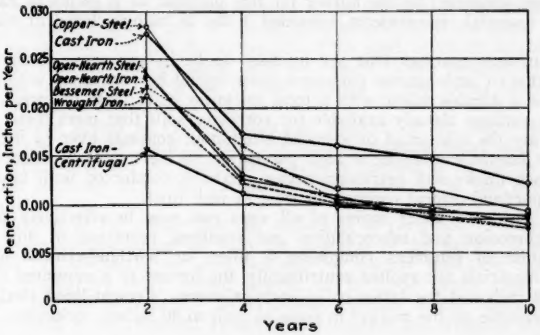


FIGURE 2.

As a result of the work centered at the National Bureau of Standards from thousands of test pieces that have now been exposed for ten years in many different types of soil, considerable progress has been made in the study of soil factors and in correlating these factors with the degree of corrosion. At present, however, it is impossible to determine with certainty the economic justification of any type of coating practice. In other words, coating practice is not yet on a strictly scientific basis. However, if this work continues (and it would be a distinct loss if it should now be dropped) it will quite likely become possible to make a reliable estimate of the value of certain coatings after an examination of the soil and the locality in which the pipe is to be laid.

ESSENTIAL REQUIREMENTS OF UNDERGROUND COATINGS.

The most essential requirements of coatings for use underground are:

(1) Material used must not be attacked by solutions in soil.

(2) It must permanently prevent contact between the soil and the pipe and exclude water, or a protective layer of inhibitive solution must be maintained between the soil and the metal. As bare pipe nearly always fails by pitting in corrosive soils, the object should be to eliminate as far as possible differences of potential on the surface of the metal, since these are the main causes of local corrosion. As these variations in potential are due mainly to variations in the soil, contact between the soil and the metal should be prevented at all points. The soil water itself is not usually very corrosive, except where local variations in concentration of dissolved substances occur.

(3) In the case of bituminous coatings, the layer next to the metal should adhere well to the metal and be made relatively soft so as to resist cracking, but the outer layer should be hard enough to resist soil stress. Distortion by soil stress is the main cause of the destruction of bituminous coatings and should be prevented by interposing a rigid and durable shield between the coating and the soil where the coating material is not sufficiently rigid in itself to prevent this action. A thin stiff rolled-steel sheet formed to the shape of the pipe in semi-circular sections and well coated, or a spiral wrapping of celluloid seems to serve fairly well for this purpose. Where the steel sheathing corrodes it reduces the corrosive matter in the adjacent soil. A single coat of vitreous enamel on each side of a thin steel sheet has been suggested by the author for this purpose, as it seems to have most of the essential requirements, provided it can be manufactured at reasonable cost.

Bituminous coatings that are intended to be in any sense permanent for protection of pipe against corrosive soils, should be reinforced with an inert filler or a durable fabric with a total thickness of at least ⅛ inch.

The coatings already available for corrosive soils that meet these requirements are the reinforced or shielded bituminous coatings over ⅛ inch thick; asphalt mastic, containing a high percentage of inert mineral material, over 3/16 inch thick; stiff petroleum-grease mixtures reinforced with fabric; and dense portland-cement concrete at least ⅜ inch thick.

The inside of water mains of all sizes can now be effectively protected from corrosion and tuberculation and resulting reduction in flow by the application of bitumens containing a filler, or portland-cement mixtures. These materials are applied centrifugally, the former to a minimum thickness of 1/16 inch and the latter to ⅛ inch or more. Cement-lined steel pipe is now available on the market in sizes ¾ inch to 30 inches, inclusive.

CONCLUSIONS.

Prevention of corrosion is an economic problem, the practical solution of which requires the cooperation and best judgment of chemists and engineers, with full knowledge of all the factors, especially those that control the reaction in each particular case.

A fundamental study of metal surface reactions and of the bond existing between metals and films, and various means for forming these and other protective layers (solid or liquid), should assist future improvements in protection of metals from deterioration. Practical means of prevention of corrosion have been worked out in most cases, and the selection of the best method has now become largely a question of economics. The use of a great excess in thickness of metal is usually wasteful and particularly objectionable where saving in weight is desirable. Rust-resisting steels often give the answer; frequently it is a matter of controlling the environment by elimination of the cause; but in most cases the solution of this world-wide problem is found in the proper application or building-up of an adequate protective layer on the metal.—"Mechanical Engineering," June, 1935.

THE CORROSION PROBLEMS OF THE NAVAL ARCHITECT.

By Dr. W. H. Hatfield.

(Institution of Naval Architects; April, 1935.)

This subject can be dealt with best by considering the corrosion of steel from two angles:

Part I.—The behavior under marine conditions of the steel normally used for the construction of ships.

Part II.—The metallurgical progress achieved in producing steels resistant to such conditions.

PART I.—STEELS NORMALLY USED.

The last century saw the invention and development of the Bessemer and open-hearth processes which, coupled up to the highly developed blast furnaces on the one hand and to the plate mills on the other, resulted in rendering available the necessary tonnage of mild-steel ship plates and steel sections at such an economic value as to render possible the shipbuilding achievements of the last few decades. This mild steel, while having great ductility, corrodes very rapidly unless adequately protected. The preservation of the ship's hull with the passing of time is the function of the efficiency of the paint applied, and not that of the intrinsic properties of the steel. If the reservation is made that the characteristics of the steel, or of its surface, may govern the efficiency of the paint, still we return to the position as stated in the last sentence. The whole of this problem is now being investigated by the Corrosion Committee.

There would appear to exist considerable support for the impression that the corrosion of ships' hulls is more frequent in severity today than it was 20 to 40 years ago, and the Committee therefore decided in its marine corrosion work to study first this question of the corrosion of the outside of ships' hulls.

It is obvious at the outset that the resistance to corrosion of ships' hulls must be very closely bound up with the behavior of the paint. If painting is carried out efficiently on plate surfaces of normal condition, and with sufficient regularity, there would appear to be little doubt that serious corrosion will be prevented. Nevertheless, opinion seems fairly general that what used to be regarded as sufficiently careful and frequent treatment in this respect does not give as uniformly satisfactory results in all cases as it did some years ago. If this opinion is a true estimate of the state of affairs, various possible causes must be considered. For example, it has been suggested as possible that differences arising in the manufacture of the steel, either in the actual raw materials and melting process, or in the rolling, may exert an appreciable influence. On the other hand, it is suggested as probable that the reduction in total weathering time between leaving the rolls and the first painting, which has occurred in recent years, has had an appreciable influence. Clearly, too, painting procedure and the nature of the paints used require careful consideration; while the care given to maintenance during service, including frequency of dry-docking, etc., can have a very considerable effect on the life of the plates. All these factors require careful study, and tests carried out should include not only a study of a large number of relatively small test-samples representing a wide variety of conditions and exposed in stationary racks at various ports, but also careful observation of the behavior of actual ships' plates under normal service conditions.

The capacity of resistance to corrosion of rivets is a further matter requiring careful study. It has been clearly indicated in the information received by the Committee, as well as by such papers as that by J. Montgomerie and W. E. Lewis, and that by W. Bennett, how serious a matter rivet corrosion may become. Apart from other questions, the relative value of wrought-iron, as against steel rivets, when both are of sound quality is a matter which is still in doubt, and on which definite information would be welcomed.

Mention of wrought iron brings to mind the old controversy as to the relative merits of iron and steel for hull construction generally. Several interesting cases of longevity of wrought-iron ships have been brought to the

Committee's notice. At the same time, evidence is not lacking that serious troubles have been experienced with wrought iron while very excellent service has been obtained with mild steel. Thus the question of the relative merits of the two materials is still one requiring study.

In some cases very abnormally rapid corrosion occurs, and information regarding a number of such cases has been sent to the Committee. Sometimes the intense corrosion has probably been caused by excessive mechanical damage to the paint. Other possible causes of such troubles are electrical leakage from cables or equipment on the ship, pollution of the water in which the ship lies, insufficient painting, and the flaking off of mill scale, together with the attached paint, due to damage, stress, or unsatisfactory weathering.

The question of the influence of the rolling scale is undoubtedly one of great importance. Thus a scale that has already been partially undermined by corrosion during weathering, if not removed by scratch brushing before painting, will in all probability flake off in time, carrying the paint with it. Pickling, of course, prevents any such action, but it is an expensive process, and, although standard practice for the Admiralty, is rarely used in mercantile work. The degree to which de-scaling by weathering should be permitted is, therefore, a matter of great interest, demanding careful study, which should include investigation of the influence of different rolling treatments, etc., on this weathering.

Numerous cases have been reported to the Corrosion Committee of corrosion which has resulted from mechanical damage to the paint.

The evidence is not always so complete as would be necessary to fix definitely the reason for the whole of the abnormal attack noted, but in a number of other cases mechanical damage from blows, or attrition by cables, etc., has almost certainly been a contributory, if not a main, cause.

Other interesting causes of mechanical damage to paint and plates which have been reported to the Committee are:

(*a*) Scouring by ashes from ash ejectors, which may reduce the life of the plates from the normal 20 to 25 years to 8 to 10 years. Naturally, after removal of the paint, both corrosion and erosion of the plate can occur in this case.

(*b*) Scouring action of sand in suspension in fast-moving water, as in the case of the Hooghly River. Here, again, both corrosion and erosion of the plates can occur. One firm has stated that a reduction of as much as ⅛ inch can occur on bow plates in five to six years, bringing a 5/16 inch plate well within the scrapping limit.

(*c*) Continual flow of scupper discharge over plates. This, again, is quoted as reducing the life of plates 8 to 10 years, from the normal 20 to 25 years.

Trouble from defective paint coatings can arise from the impossibility of proper painting, as in the case of the keel plates where the presence of the keel blocks prevents complete repainting. Extra heavy plates are, of course, provided here to allow for the wastage, which may amount to as much as ⅜ inch in 20 years.

The old question as to the possible superiority of acid or basic steel has arisen in connection with some of the reports received. Thus, for example, one company, after describing a case of abnormal corrosion, goes on to state that the vessel was built from acid steel. On the other hand, another company is of the opinion that basic steel is more prone to corrosion. There are so many possible causes which can contribute to marine corrosion that this question as to the relative merits of the two types of steel can only be settled by a very careful scientific study.

One company has reported that they have found corrosion of the heads of iron rivets in the countersink, where no corrosion was found on the steel plates. This attack occurred at the bilges, where the rivets were subjected to more stress than at other parts of the steel plating.

Finally, an interesting report may be quoted as illustrating various corrosion troubles arising from too long delay before repainting, thus making it possible for damaged paint to permit appreciable corrosion of the underlying plates.

In addition to collecting information from various sources, the Committee has, in a number of cases, carried out through its members first-hand examination of ships, and in two cases has been able to arrange for the building in of a number of plates of completely known history treated in various ways as regards rolling, removal of scale, and painting.

As matters stand at present, it is considered that, since the effectiveness of the paint is largely affected by the condition of the surface of the plates at the time of painting, considerable attention can with advantage be given to the effect of the amount of scale, its nature, and also to the effect of freedom from scale. The author believes it to be true that in the old days hulls were launched at times without initial painting, and that the saline estuary water pickled away the scale, and in such cases made the painting a more efficient job.

Scale may be removed by pickling or by sand-blasting, but the cost of such procedure would add very materially to the cost. The Committee's experiments also indicate that plates oiled while hot effectively retain their paint; again expense is entailed.

If rolling temperatures are modified to produce a desirable type of scale, the output of the mill is seriously affected, and thus in this direction again additive cost would be entailed.

In the very competitive business of shipbuilding any introduction of more costly technique is undesirable, but it is clear that the shipowner and shipbuilder should weigh the added maintenance charges which are occasionally encountered with bad cases against the cost of procedure, the result of which appears to give greater hope of eliminating such cases.

PART II. STEELS RESISTANT TO MARINE CONDITIONS.

Plates and sections of mild steel have initially invariably an oxide, and generally a scale, surface. The chemical composition and physical characteristics of the scale and of the underlying oxide layer are variable and unreliable. They are mechanically fragile, and are usually not impervious to aqueous solutions, hence the difficulties discussed in the preceding section. The underlying metallic steel is readily attacked by sea-water. What is the reason for this action of sea-water? The details of the mechanism are still subject to differences of opinion, but, broadly speaking, the process is an electrolytic one, the surface of the mild steel containing potentially anodic and cathodic areas which, in the presence of such an excellent electrolyte as sea-water, lead to solution of iron at the anodic areas. The only effective manner in which this process can be arrested is by applying coatings of paint. This paint, although permeable to some extent, offers a very high resistance to the action of the sea-water, and thus reduces the attack to negligible proportions. Unfortunately, paint deteriorates with time, and it is liable to suffer seriously from mechanical damage, while periodic repainting is costly and often inconvenient. It is therefore very desirable that steels should be available which will not suffer a serious corrosion when the unprotected metallic surface is exposed to marine conditions. The last 20 years has seen great progress in this direction.

TABLE I.—ANALYSES OF TYPICAL STEELS.

Steel.	C.	Si	Mn.	S.	P	Ni.	Cr.	W.	Ti.
Mild-steel plates and sections ... .(*a*)	0·10	0·04	0·47	0·028	0·035	0·08	—	—	—
(*b*)	0·16	0·09	0·62	0·035	0·031	0·13	—	—	—
(*c*)	0·23	0·04	0·65	0·034	0·038	0·14	—	—	—
(*d*)	0·45	0·13	0·65	0·024	0·034	0·10	—	—	—
Heavy mild-steel forgings	0·10	0·12	0·35	0·035	0·032	0·10	—	—	—
Stainless steel	0·27	0·35	0·26	0·025	0·020	0·25	13·1	—	—
Austenitic chromium-nickel steel.....(*a*)	0·12	0·55	0·30	0·025	0·015	8·15	18·2	0·63	0·65
(*b*)	0·11	0·25	0·26	0·024	0·019	12·7	12·8	—	—

TABLE II.—TYPICAL MECHANICAL AND PHYSICAL TESTS.

Steel.	M.S., tons/sq. in.	El. Lim. taken as 0·50 per cent. Proof Stress.	Y.P., tons/sq. in.	El., per-centage on 8in.	El., per-centage on 2in. × 0·564in. diam. or thickness.	Red., Area per cent.	Izod Impact.	Fatigue Limit.	Magnetic Properties.	μ for H = 50.	Thermal Expansion per 1° C. over 20/400° C.
Mild-steel plates and sections :											
(*a*) 2-in. angle	26·5	15·2	19·2	26	31	63·5	48	±13	Highly magnetic	350	0·0000136
(*b*) 6-in. angle	29·4	17·5	20·6	26	30	61·9	50	±14	Highly magnetic	320	0·0000135
(*c*) 1¼-in. plate	32·0	15·9	17·1	23·5	27·5	50	25	±14·5	Highly magnetic	290	0·0000133
(*d*) 1¼-in. plate	39·6	19·5	20·7	15·5	20	25·4	7	±16·5	Highly magnetic	270	0·0000130
Heavy mild-steel forgings	25·5	12·5	14·0	—	33	65	30	±13	Highly magnetic	350	0·0000136
Stainless steel 3in. diam.	48	28·0	35	—	26·5	60	45	±21	Magnetic	270	0·0000115
Austenitic stainless steel : CrNi 18/8 plate ...	47	14·5	21	36	50	54	100	±17·5	Practically non-magnetic	1·7	0·0000178
3-in. diameter bar ...	46	14·0	22	—	49	62·6	100	±17·6	—	—	0·0000178
Austenitic stainless steel, CrNi 12·5/12·5, 1¼-in. bar	35	10·5	16	—	64	73	100	±15	Practically non-magnetic	1·01	0·0000184

Space does not permit a detailed survey of the experimental developments, but it may be stated that the addition of about 13 to 14 per cent chromium to steel produces, as established by Brearley, the steel from which a stainless knife could be made.

In Tables I and II will be found the comparative analyses of the ordinary shipbuilding steels and of the "rustless" steels, together with the essential comparative data concerning their mechanical and physical properties.

Although metallurgical research has made such progress that rustless steels are now available, it must be recorded that the addition of large percentages of chromium and nickel, coupled with the fact that at this stage only a small tonnage of steels is produced and in many diverse forms, results in the materials having only limited application, the cost of production at the present time being considerable. Nevertheless, they are being applied in many directions, invariably where the unprotected metallic surface is desired, or where particularly onerous service has to be met.

Interesting examples may be stated.

One important place where corrosion and consequent play occur is in the rudder-post gudgeon. The mild-steel gudgeon bushes and the mild-steel bottom gudgeon cap might well be replaced with rustless castings. The author has learned with great satisfaction that an austenitic rustless-steel gudgeon bush fitted to the single gudgeon of the *Aquitania* shows practically no deterioration or sign of wear after six years of service. This is a valuable result. Thus, for rudder parts, propellers, keel bolts, stern bolts and protection plates, we have here a very useful material.

As materials for turbine blading, the high chromium stainless steel, the high nickel-chromium steel, and the high chromium-nickel are each finding their niche according to the onerousness of the particular conditions.

The advent of the superheater brought with it the problem of supports for the tubes, and, so far back as 1921, it was found that mild steel was useless when used for the service type of superheater support. It is now universal practice to use a high chromium-nickel steel to which tungsten has been added, such as H.R. Max, Pyrista and Era, and flame baffles are constructed of the same material. Incidentally, the recently converted *Asturias* and *Alcantara*, using Johnson boilers, have supports made of steels. The increased use of preheated air caused the M.S. combustion tubes to fail, and cast H.R. Max tubes and palm supports have given great service under the arduous conditions. Exhaust valves for Diesel engines are being effectively produced in similar new steels. In other words, recent metallurgical progress is supplying interesting solutions of many technical difficulties associated with marine engineering development.—"The Shipbuilder and Marine Engine-Builder," April, 1935.

WHAT EVERY BOILER OPERATOR SHOULD KNOW ABOUT SUPERHEATERS.

By Joseph Waitkus, Engineer, the Superheater Company, New York.

Many boiler operators appear to be of the opinion that the superheater can function successfully under the same conditions as the boiler heating surface. For this reason, superheaters frequently are abused and neglected, with the result that the life of the equipment is greatly reduced and the maintenance cost increased considerably. Such conditions can be avoided if a few points of difference as to boiler and superheater operation are borne in mind and reasonable care is exercised in the care of the superheater.

The boiler contains tubes of relatively large diameter, filled with water and absorbing heat from the gases of combustion. The superheater, on the other hand, consists of a large number of small-diameter tubes filled with vapor and also absorbing heat from the hot gases. In the boiler tubes the water circulates naturally by its change in density as it absorbs heat. So long as there is water in the boiler drum, and there are no dead pockets, this circulation will continually remove the heat as it is conducted through the walls of the tubes. The natural circulation of the water and the absorption of heat by this water combine to protect the boiler heating surface. Should the boiler, for some unforeseen reason, be isolated suddenly from the main steam line, the boiler surface still is protected by the natural circulation of the water in the tubes. The safety valve, of course, will relieve the excess steam, but a continual supply of feed water is injected, which maintains circulation as under normal operation on the line.

In the superheater a different condition exists. The circulation of steam through it depends entirely upon the pressure difference between the inlet and the outlet. This pressure drop is a direct function of the steam velocity which, in turn, depends upon the quantity flowing and the internal area of the tube. It can be said, therefore, that the proper combination of steam quantity and velocity provides the protection for the superheater. Since superheaters are installed between the boiler outlet and the steam main the steam flow through the units is assumed to be continuous. If the stop valve be closed, no steam will flow through the superheater units, and both the steam velocity and the pressure drop become zero because the inlet and outlet ends of the superheater are at the same pressure. It is not difficult to visualize what will take place under such a condition. With no steam flowing to remove the heat as it is conducted from the outer to the inner surface of the tube the temperature approaches that of the gases flowing past the superheater. If this temperature is higher than the safe temperature for the tube material, failure may result from burning or oxidation of the material.

It may be concluded, therefore, that circulation of steam in sufficient quantity through the superheater units is of vital importance, particularly when it becomes necessary to start up or shut down the boiler. For a considerable period during both of these operations the boiler stop valve is closed and no steam passes through the superheater to the steam main, although hot gases are passing over the superheater surface. To meet this condition the superheater is provided with a suitable drain valve on both the saturated- and the superheated-steam headers. The purpose of the drain valve on the saturated-steam header is to remove any condensed steam in the header and thereby permit free movement of steam from the boiler to the superheater units. This valve is closed as soon as the condensate is drained away. The drain valve on the superheated header when open will permit circulation to take place through the superheater units which protects them while the boiler is gradually brought up to the operating pressure.

Too frequently operators give only a half turn to the drain valve, having in mind the saving of steam. While some circulation is set up thereby, it may not be enough to protect the superheater. The size of the valve is selected with the express purpose of passing sufficient steam to get the desired circulation, and boiler operators should take advantage of it. It is far better to lose a few dollars in steam than several hundred dollars in superheater tubing. The drain can be directed to a sewer or sump through piping of the same diameter as the valve. It should never be connected to a trap or any contrivance which might hinder the free flow of steam. When the boiler has reached the operating pressure and is on the line, then and only then should the drain valve be closed. When the boiler is being taken off the line the drain valve should be opened again as soon as the stop valve is closed, and remain open until the boiler is placed on the line again.

In some plants it is necessary to bring the boiler on the line far more rapidly than is normal practice. This requires a large release of heat in the furnace in order to generate steam rapidly. To meet this condition plenty of steam circulation and velocity is necessary through the superheater units. Drain valves larger than those usually furnished will aid the situation. Under extreme conditions, as with a radiant-type superheater, it may be necessary to furnish steam from another boiler in sufficient quantity to keep the superheater cool until the boiler reaches the operating pressure and is placed on the line.

The boiler operator will find it very convenient to insert a thermometer in a thermometer well, located in the superheated-steam header, and, as the boiler is building up pressure, make periodic observations of the steam temperature. This will permit close control of the drain valve so as to keep the temperature down to a safe limit of about 800 F. and at the same time drain only as much steam as is necessary to keep within this limit.

Operators sometimes raise the question as to why it is necessary to have the safety valve at the superheater outlet and, furthermore, why the safety valve should be set from 3 to 5 pounds below the setting of the boiler safety valve. These questions have a bearing on the circulation of the steam through the superheater units. When the boiler safety valve opens there is a rush of steam to the atmosphere and, since the pressure difference is greater through the safety valve than through the superheater units, practically the entire steam flow is directed through the safety valve. The superheater, therefore, is robbed of the steam needed to protect it. To remedy this condition, a safety valve is provided at the superheater outlet and is set from 3 to 5 pounds below the boiler safety-valve setting. The superheater safety valve, therefore, will blow before the boiler safety valve, which means that the steam, in passing from the boiler to the atmosphere, must pass through the superheater and thereby protect the units. Operators should be cautioned not to change the relative settings of the boiler and superheater safety valves, once they are set properly.

Now consider the gases flowing over the external surface of the superheater. The temperature of these gases frequently will determine, to a large extent, the degree of care necessary in operating the superheater. For instance, a radiant superheater located directly in the furnace will require particularly close attention. The gas temperatures are sometimes in the neighborhood of 2500 F. Such temperatures would be most detrimental to any tube material used today in superheater manufacture, if steam flow ceased in the superheater units even for a short period. If, on the other hand, we consider the convection type of superheater, illustrated in Figure 1, it will be noted that there is a large amount of boiler heating surface ahead of the superheater. This surface reduces the temperature of the gases considerably so that in passing over the superheater surface they are relatively low in temperature, compared to the radiant superheater. Cases have been reported, with superheaters of the type shown in Figure 1, where there has been occasion to disconnect the superheater units from the header and, without removing the units, to continue operation of the boiler for quite a period. When these units were removed at some more convenient time they were found to be in good condition and ready for further operation. These, however, are exceptional cases and are cited simply as an illustration of the protection a superheater can get from the boiler heating surface ahead of it.

One of the greatest sources of trouble in superheater installations is flame impingement. If located in a region where flames can impinge on the tube surface, failure will occur sooner or later in spite of what may be considered as continuous and normal steam flow. It is not difficult to explain why this is so. The flame is hot enough to burn the tube surface and form a thin

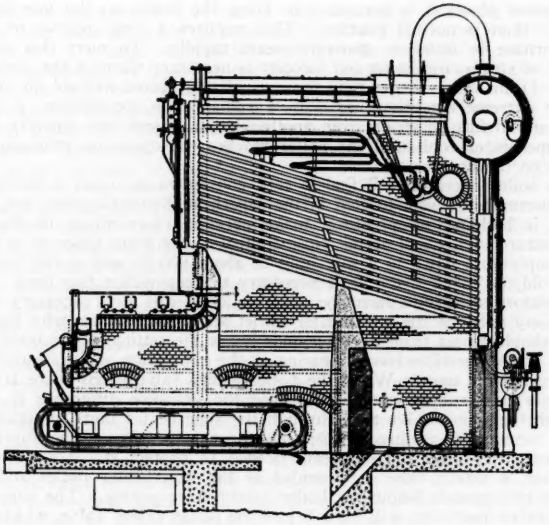


FIGURE 1—TYPICAL CONVECTION SUPERHEATER IN CROSS-DRUM BOILER. THE LARGE AMOUNT OF BOILER HEATING SURFACE AHEAD OF THE SUPERHEATER PROTECTS THE LATTER.

layer of oxide. The oxide then is worn away by the erosive action of the flame as it sweeps over the tube surface and uncovers the bare surface of the tube. The continual oxidation and erosion of the tube material soon wears the tube section down until it is too thin to stand the pressure and failure results. The time it takes to cause failure will depend upon conditions in the furnace combined with the quantity of steam flowing through the units and its velocity. The views in Figure 2 illustrate a typical tube failure from flame impingement. Note the thinned cross-section on the failed side, also the thin layer of oxide broken away in places on the tube surface. If flame impingement cannot be avoided the superheater surface must be covered with a protective material which can be renewed from time to time.

The point to bear in mind in connection with flame impingement is the fact that it is due entirely to furnace conditions. Proper adjustment of burners and combustion, and close observation of flame length, with particular notice of its proximity to the superheater, will aid materially in eliminating an important source of trouble. In giving furnace operation data for the design of a superheater, too frequently information regarding flame conditions is omitted, with the result that trouble arises because provisions were not made to protect the superheater properly, nor to install it in a more advanageous position.

On the subject of furnace conditions another very common source of trouble is the so-called "secondary combustion." Superheater design is based on the assumption that combustion of the fuel is completed in the furnace and that only the products of combustion pass over the surface of

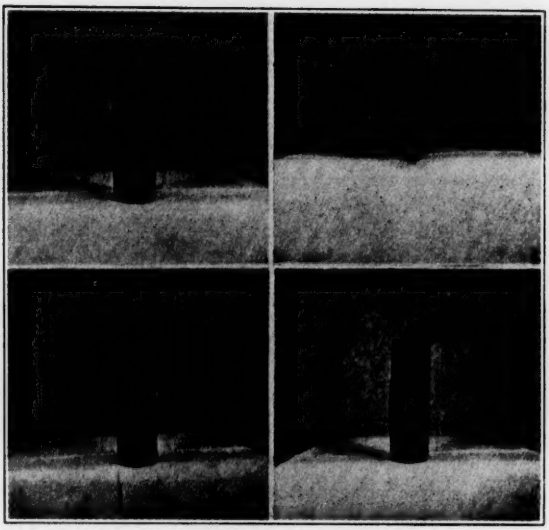


FIGURE 2—A TYPICAL SUPERHEATER TUBE FAILURE CAUSED BY FLAME IMPINGEMENT.

the superheater. This applies to the convection superheater. The radiant type, of course, depends largely upon the heat radiated from the flame. A properly designed furnace is provided with sufficient volume in which the fuel is burned completely, so that only the resulting gases of combustion pass over the heat-absorbing surfaces. If the furnace is deficient in volume, this deficiency will be made up in the gas passes where the fuel will continue to burn, if sufficient air is present to support combustion. Assuming the furnace has sufficient volume, if the proper amount of air is not provided for complete combustion in the furnace, the unburned gases will pass through the boiler and burn at points where air leakage through the setting will furnish sufficient air for further combustion. Applying these two conditions to a superheater located as illustrated in Figure 3, there will exist, in the first place, a condition almost analogous to flame impingement and, in the second place, the superheater is being robbed of the gases it normally would get under proper furnace operation. In the first case, surface oxidation and erosion may be expected, with the additional evil of a very high degree of superheat due to temperatures on the external surface being higher than anticipated; and in the second case, the superheat is likely to be low because of insufficient gas flow and low gas temperatures.

Slagging is well known among boiler operators, but its effect on general boiler performance frequently is overlooked. Slagging is the result of furnace operation which creates a furnace temperature so high that the ash becomes molten or plastic. The particles of ash in this state are carried up into the boiler tubes where they solidify and cling to the relatively cool tube surface. The accumulation progresses until a fairly thick layer is deposited on the leading surface of the tubes. Slag has a very low coefficient of heat transfer, hence surfaces covered with it are very materially reduced in their efficiency as heat absorbers.

The effect of slagging on superheaters will depend largely upon the type of boiler used and the location of the superheater. For example, in Figure 1 the superheater is well protected by the boiler heating surface. There is little danger of slag accumulating on the superheater surface. Slag accumulations on the boiler heating surface, however, will cut down their heat pick-up enough to cause the superheater to be subjected to gas temperatures higher than anticipated, causing high superheat. On the other hand, a superheater located as in Figure 3 will suffer from slag deposits, causing low superheat, unless care is exercised. Superheaters frequently are located, as indicated in Figure 4, where again a reasonable amount of boiler surface is ahead of the superheater. Results similar to those from the arrangement in Figure 1 can be expected from this type of installation.

Boiler operators, therefore, should pay strict attention to the furnace temperature and adjust furnace conditions so as to keep the temperature of the gases entering the boiler heating surface below the fusion temperature of the ash. Any amount of slag is a detriment and a serious menace to good operating results.

Few operators realize that moisture content of the steam or "carry-over" plays a most significant part in superheater operation. The fundamental purpose of a superheater is first to remove whatever moisture may be contained in the saturated steam, then to raise the temperature of the dry steam to a predetermined value. If the moisture content is excessive the superheater actually is functioning as a boiler and the heat which otherwise would be used to raise the temperature of the steam is being utilized to evaporate water. This in itself is a serious condition, but the most important problem is the formation of deposits on the inner surface of the tubes due to this excessive moisture. Boiler feedwater, whether treated or not, contains certain proportions of chemicals which form a deposit as the water is evaporated.

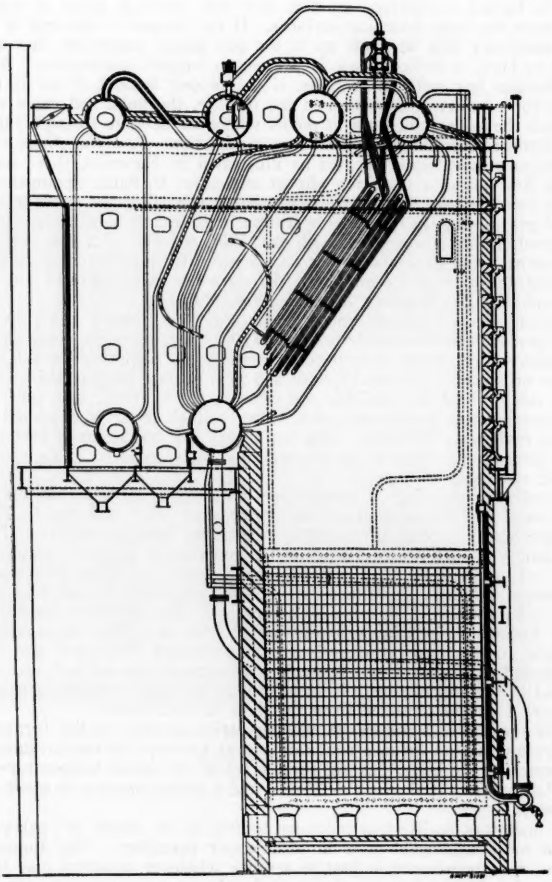


FIGURE 3—AN INTERTUBE SUPERHEATER IN SEMI-VERTICAL BOILER MAY SUFFER FROM SLAG DEPOSITS OR EXCESSIVE GAS TEMPERATURES, UNLESS CARE IS EXERCISED IN OPERATION. LOCATION OF BURNERS NOT SHOWN.

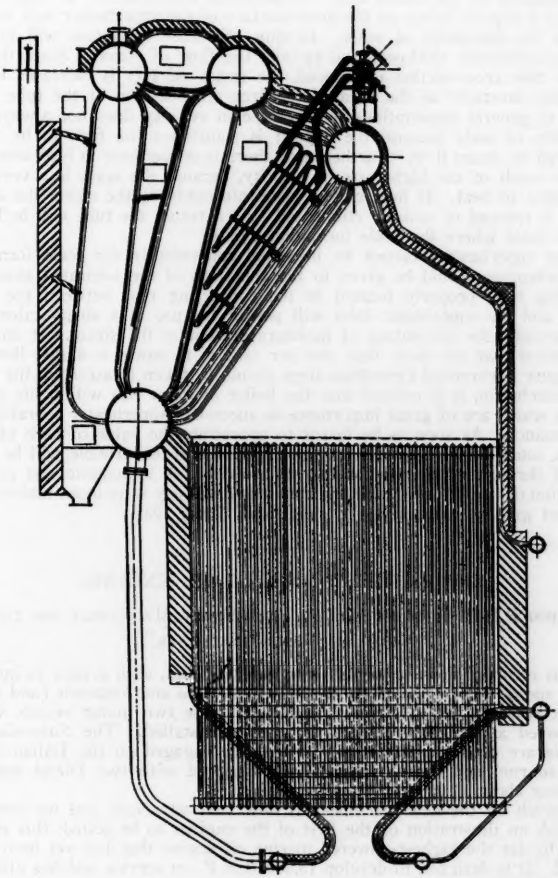


FIGURE 4.—SUPERHEATER ENTIRELY BEHIND TUBE BANK OF SEMI-VERTICAL BOILER. THE BOILER TUBES PROTECT THE SUPERHEATER. LOCATION OF BURNERS NOT SHOWN.

These same chemicals are contained in the moisture carried over into the superheater with the saturated steam. As the moisture is evaporated under the influence of the rather high temperatures characteristic of superheating surface a deposit forms on the inner surface of the superheater unit and thus begins the formation of scale. In time the scale formation will build up to a considerable thickness and restrict the flow of steam. Since the area of the tube cross-section is reduced, the steam velocity is increased because it varies inversely as the area of the free cross-section of the tube. Contrary to general expectations the increase in velocity does not always clear the tube of scale because the deposit is sometimes too firm to be readily removed by steam flow. Furthermore, there is no increase in heat absorption as the result of the higher steam velocity, because the scale is a very poor conductor of heat. If heat cannot penetrate easily to the steam the cooling effect is reduced or entirely eliminated. As a result, the tube will be burned at the point where the scale formation exists.

Since superheaters cannot be inspected conveniently for scale formation, close attention should be given to the condition of the saturated steam. A sampling tube properly located in the connecting pipe between the boiler outlet and the superheater inlet will permit the use of a steam calorimeter to determine the percentage of moisture present in the steam. It has been established that not more than one per cent of moisture is a safe limit. If this figure is exceeded immediate steps should be taken to ascertain the cause.

In conclusion, it is evident that the boiler operator has within his control factors which are of great importance to successful superheater operation and performance. As soon as he learns to appreciate the value of each of these factors, much better operation can be expected and less trouble will be experienced throughout the life of the superheater. A systematic and periodic inspection of operating routine and conditions will pay in reduced maintenance cost and greater reliability.—"Combustion," June, 1935.

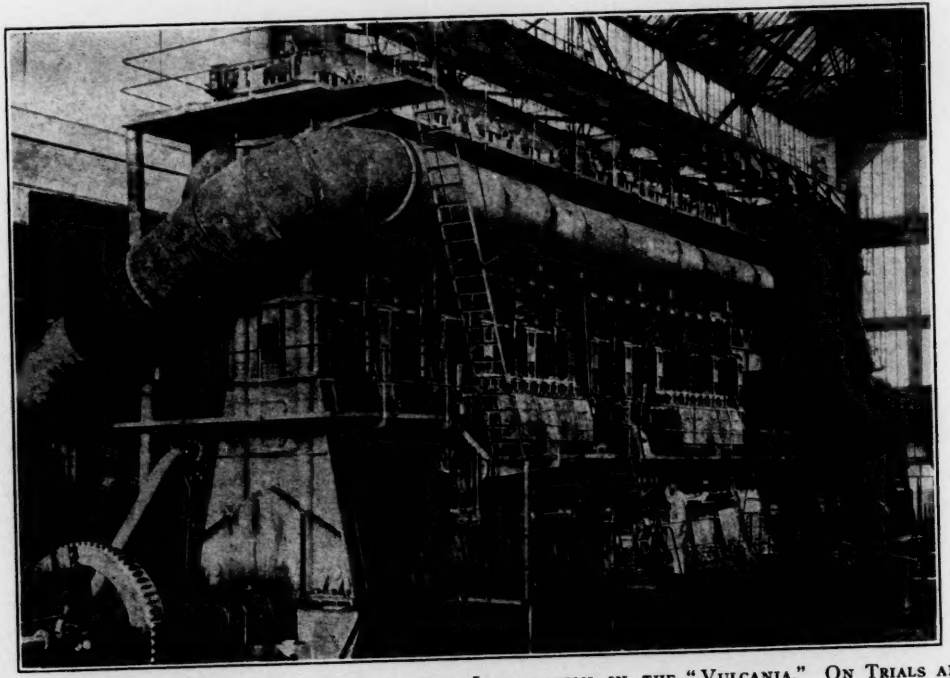
THE LARGEST MARINE OIL ENGINE.

24,000 B.H.P. to 26,000 B.H.P. Twin-Screw Machinery for the "Vulcania" and "Saturnia."

It was recorded in this journal some time ago that, with a view to increasing the speed of the Cosulich motor liners *Vulcania* and *Saturnia* (and solely for that reason), the existing machinery of these two motor vessels would be removed and higher-powered oil engines installed. The *Saturnia* and *Vulcania* are of about 24,000 tons gross, are engaged on the Italian-South American run, and are at present each equipped with two Diesel engines, developing about 17,000 B.H.P. in service.

The work on the new machinery is now well advanced, and we are able to publish an illustration of the first of the engines to be tested, this representing by far the highest-powered marine oil engine that has yet been constructed. It is designed to develop 13,000 B.H.P. on service and has attained 16,500 B.H.P. on trial.

In the *Vulcania* two Fiat double-acting two-stroke engines are to be installed (built by the Fiat Stabilimento Grandi Motori, Turin), but in the *Saturnia* two Sulzer-type double-acting two-stroke motors constructed by the Cantieri Riuniti dell'Adriatico will be fitted. The specification calls for an output of 24,000 B.H.P. to 26,000 B.H.P. per ship on service, but a very substantial overload capacity is available. The length of the ships will be increased somewhat and new bows constructed, which it is anticipated will bring the speed up to at least 21 knots.



ONE OF TWO 13,000 B.H.P. FIAT ENGINES FOR INSTALLATION IN THE "VULCANIA." ON TRIALS AN OUTPUT OF 16,500 B.H.P. WAS DEVELOPED. THE SEPARATE DIESEL-ENGINE-DRIVEN SCAVENGING PUMP IS SEEN ON THE RIGHT.

Dealing first with the Fiat engines to be installed in the *Vulcania*, each has ten cylinders with a diameter of 750 millimeters and a piston stroke of 1250 millimeters, while the normal speed in service will be 128 R.P.M. This is practically the same speed as that of the existing engines. The designed output is 13,000 B.H.P., and, as already recorded, in the bench trials the first engine developed 16,500 B.H.P.

DIESEL-ENGINE-DRIVEN SCAVENGING PUMPS.

In the existing arrangement of machinery (of the four-stroke double-acting injection type) in the *Vulcania* the air for the injection of the fuel in the main engines is supplied from two separate 1350 B.H.P. Diesel-engine-driven compressors. The new propelling engines are of the airless-injection type, hence the compressor sets are not required, and in their place two other units will be installed, each comprising a Fiat single-acting two-stroke 1300 B.H.P. Diesel engine directly coupled to a reciprocating scavenge pump. This will supply the scavenging air for the main propelling engines (one unit being sufficient in the normal way), while there will also be a stand-by scavenging blower driven by a steam turbine. In addition, a steam turbo-generating set of 400 Kw. will be installed, and all of the steam necessary will be supplied from two exhaust-gas boilers. These boilers will also provide the remaining steam needed for ship's services, and it is believed that no oil firing will be needed while the ship is under way. The economy thereby effected will undoubtedly be considerable.

The present installation for cooling water and lubricating oil will not be changed except that, in view of the higher power, the output will be increased.

It is anticipated that the total increase in propelling power will be nearly 50 per cent above that at present installed. Despite this fact, the new propelling machinery will be installed in the present engine-room, and this applies also to the *Saturnia*.

In the new design of Fiat double-acting two-stroke engine, which follows the long experimental work carried out on a 2000-horsepower cylinder, the main features of the single-acting engines have been retained so far as possible. Steel is largely employed in the construction, the cylinder heads being a single steel casting in each case, while the upper and lower liners are of steel, capable of expansion in the center. Each jacket has the two ends of cast steel, while the center portion is of cast iron. The jackets are carried on columns, which are bolted to the bedplate, and as all the cylinder jackets are bolted together they form a rigid beam, which strengthens the upper portion of the engine. The bedplate is in sections bolted together and having transverse bridges to carry the main bearings. The shells of the bearings are of steel, lined with white metal, and can be removed with the crankshaft in place.

Into each steel cylinder liner is pressed a cast-iron bush, so that the replacement of the liner is neither an expensive nor a long proceeding. The liners are held in place by the cylinder heads, and the scavenging and exhaust ports are arranged in the cylinder jacket.

The manner in which the framework is built up is readily visible in the illustration of the engine reproduced on the next page. The through bolts extend from the top of the cylinders to the transverse bridges of the bedplate, and the nuts at the tops can be seen. The cast-iron portions of the framework are, therefore, relieved of tension stresses, and rigidity is augmented by bolting the crosshead guides to adjacent columns.

The crankcase is enclosed by a cover forming an oiltight sump, and the crankshaft is of the semi-built-up type, bored throughout its length for the passage of lubricating oil to the bearings.

The pistons are formed in three sections, the upper and lower heads being of cast steel and the center portion of cast iron. A tube surrounds the piston rod to protect it from the gases at the maximum temperature, and in the lower cylinder cover is a stuffing box, through which the piston rod passes. There is another stuffing box at the top of the crankcase.

Oil is employed for cooling the pistons. It is delivered to the bottom piston head through the annular space between the piston rod and the tube around it. The oil passes up the piston to the top, and is discharged through the piston rod, which has a hole bored in the center. Arrangements are made so that water cooling through the intermediary of telescopic pipes may be utilized if it is thought desirable.

A starting valve is fitted in each upper cylinder cover (not in the bottom), and while there is one fuel injection valve centrally placed in each cylinder head, two are provided in the bottom cylinder covers. All have water-cooled injectors, since it is possible that low-grade fuels may be employed. The fuel pumps are located in two groups in the front of the engine, as indicated in the illustration, and the pipes leading from the fuel pumps to the valves are of equal length.

The first engine, having completed satisfactory trials, has now been dismantled, and the second unit will probably be erected on the test bed at Turin by the end of this month.

THE MACHINERY OF THE "SATURNIA."

The two C.R.A.-Sulzer 10-cylinder engines to be installed in the *Saturnia* are also of the airless-injection double-acting two-stroke type. The cylinder diameter is 760 millimeters and the piston stroke 1200 millimeters, these being dimensions that have already been employed with Sulzer double-acting two-stroke machinery, so that the engines may be considered as standard, except that the number of cylinders is larger than has hitherto been employed.

It is extremely interesting, however, to note that in the *Saturnia*, instead of Diesel-engine-driven piston scavenging pumps being employed, electrically driven turbo-scavenging pumps are utilized. There are to be three, one of which will be a stand-by, and the necessary power will be supplied from two C.R.A.-Sulzer-engined generators each of 900 Kw., one of these being a spare.

Fresh-water cooling is employed throughout for the propelling engines. The cylinders are cast in two sections. The upper cylinder covers are of cast iron and the lower heads of cast steel.

The pistons are built up of three sections, the top and bottom being of cast steel and fitted with piston rings, while the intermediate portion is of cast iron and acts as a guide. In order to prevent corrosion of the piston rods through the cooling system, the water is discharged through the center of the rods, in which tubes are inserted to prevent contact with the rod surface. Starting air is supplied only to the lower ends of the cylinders. In the upper cylinder head is one fuel injection valve, two being located in the bottom cover, and the plungers for the individual fuel pumps are actuated from a gear-driven shaft. There are two levers for controlling the operation of the engine, one being used for starting by means of compressed air, and the other for regulating the speed of the motor. The nominal output of each engine is 12,000 B.H.P., but it is anticipated that a substantial overload will be possible.

The exhaust gases from the two engines are to be delivered to a Cochran waste-heat boiler in two sections.—"British Motor Ship," June, 1935.

STEAM PROPULSION—WHAT OF THE FUTURE?

Striking Progress Predicted.

The designer of the machinery of the *Bremen* and other outstanding vessels, and the active sponsor of the Bauer-Wach system and other important developments, urges the desirability of turning attention once again to pulverized fuel. Water-tube boilers are recommended for combination machinery ships, and remarkably low specific consumptions are suggested for future steam installations.

By Dr. Ing. Gustav Bauer.

Complying with your request I am giving you in the following notes my views on the future of marine steam propulsion, although it is difficult to exhaust this subject in a few words.

For smaller and medium-sized vessels, for powers up to approximately 7000 I.H.P. per shaft, the steam reciprocating engine will, as it has done in the past, maintain its supremacy, provided, however, that its principal shortcoming of limited expansion of the steam in the low-pressure cylinder is compensated for by the addition of an exhaust-steam turbine. This procedure has already been found to be eminently effective in more than 300 vessels. Amongst the improvements of the reciprocating engine proper, the so-called Lentz poppet-valve marine engine has proved to be rather successful. Ever since this system was used in combination with exhaust-steam turbines it has given particularly attractive performances.

TURBO-COMPOUND ENGINE.

For very small powers the turbo-compound engine, a combination of a compound reciprocating engine with an exhaust-steam turbine, should have a promising future in view of the excellent results already obtained with many small vessels.

In order to first discuss the smaller and medium-sized plants it seems to me that the most promising problem of the future will be the endeavor to replace the usual Scotch boilers with water-tube boilers, and to introduce higher steam pressures and temperatures. I do not regard it as impossible to obtain with such high-pressure, high-temperature, installations, which generate the steam in water-tube boilers and use it in combination with propelling units according to the Bauer-Wach system, when using oil fuel, a consumption of only 0.55 pound per I.H.P. per hour including propelling auxiliaries.

PULVERIZED COAL AGAIN?

In countries which produce coal, as this is the case particularly in England and Germany, sooner or later the problem of pulverized coal in connection with water-tube boilers will have to be solved in order to approach costs for the specific fuel consumption, which are at present considered unbelievably low. I consider the solution of this problem one of the most important duties of marine engineering on account of its vital importance for world shipping.

For plants of larger powers the straight turbine will remain without competition. The progress of the introduction of high-pressure, highly-superheated steam, combined with the use of oil-fired water-tube boilers, cannot be retarded any more. It will be the right thing, however, to confine ourselves to pressures of between 1000 to 1150 pounds per square inch and steam

temperatures of from 850 to 880 degrees F., partly because with higher pressures the obtainable economic advantages are too small to justify the complications in operation and the first cost, both of which increase with increasing working pressure, and partly because the utilization of higher temperatures is handicapped—if not made impossible—by the characteristics of present-day materials.

I believe that it is possible to obtain with large turbine plants, as described above, consumption figures down to 0.49 pound per S.H.P. per hour and probably even slightly less for the main propelling plant, including all auxiliaries necessary for its operation.

Modern times in many cases require an increase in speed of existing steamships. While in vessels with reciprocating engines the addition of an exhaust-steam turbine has become the acknowledged method of achieving this end, in turbine-driven ships the application of an additional high-pressure power plant, superimposed on the existing system, may be recommended.

This principle consists in the addition of a comparatively small steam turbine which is fed from a small additional boiler plant generating super high-pressure steam. The exhaust from this additional turbine is supplied, together with the steam generated in the existing boilers, to the original turbines.

DIESELS.

While the above is a short summary of my views on the future of steam propulsion, I would like to add that in a number of applications—in some of them exclusively—the Diesel engine is the superior prime mover. On the other hand, however, especially in view of the many possibilities offered by steam propulsion, it would be futile to assume that Diesel propulsion will ever replace wholly or in the majority of cases steam propulsion, especially where very large powers are concerned.

Attractions of Turbine Propulsion.

The author of "The Marine Steam Turbine" and other standard books on marine engineering, sees a wider future for turbine drive, but is cautious regarding the immediate prospects of the newest boiler types.

By J. W. M. Sothern, M.I.E.S., M.I.M.E., etc.

In these days of high-pressure steam, superheat, multiple-stage feed-water heating, air preheating, powdered coal, mechanical stoking, etc., it would appear that the limit has been reached as regards marine steam plant thermal efficiency. However high this figure may be as referred to the most recent practice, the indisputable fact remains that with Diesel machinery the figure is still higher, and occupies premier position as regards economy of fuel. In the writer's opinion, the triple-expansion and quadruple-expansion engine must inevitably give place, in the near future, to the geared turbine with high-pressure superheated steam water-tube boilers. Such an installation is better suited to compete with the Diesel. Even in the case of tramp vessels increase of speed above the old standard is now considered by shipowners to be necessary to obtain the booking of paying freights for their tonnage. The many auxiliary gears recently introduced, which have nearly all been devised with the object of effectively utilizing the latent heat energy contained in low-pressure L.P. cylinder exhaust steam, rely on a turbine to obtain this result. Therefore, the turbine is admittedly, after all, the most economical type of steam-driven prime mover for ships.

As is well known, if strict economy does not require to be seriously considered, the turbo-electric drive appears to be the ideal machinery for high-class passenger liners, where comfort and smooth running of the main machinery is of vital importance. The latest arrangement of turbines as designed by the Parsons Marine Steam Turbine Co., Ltd., for moderate powers and low-speed cargo vessels appears to be quite an attractive and practical proposition. In the opinion of the writer the only weak spot in the design is the double-reduction gearing. Single-reduction gearing would have simplified the layout and considerably enhanced the attractiveness of the scheme had this arrangement been at all possible. Doubtless, considerations of propeller efficiency necessitated the employment of double gears, although it might be suggested that propeller designers might evolve an efficient, comparatively quick-running propeller capable of being used behind full cargo ship forms.

The introduction of high-pressure, flash-type boilers, now in the experimental stage, may ultimately result in still further increase in thermal efficiency and reduced fuel consumption per I.H.P. or S.H.P. per hour. This type of boiler may, however, possess an element of serious danger in case of accident owing to the light construction of the various tubes and other fittings, the appearance of which does not instill a feeling of complete confidence in the plant to the uninitiated observer. Only sea experience can demonstrate the practical value of these new designs, concerning which we must meanwhile maintain an open mind.

As regards the question of reliability, the three or four-cylinder triple-expansion engine cannot be excelled. In the case of breakdown, repairs, alterations, or adjustments which may be required to maintain running, the engine-room staff may always be safely trusted to do the right thing, as proved by the records of many years of service. At the same time it must be admitted that the steam turbine occupies the same meritorious position as regards reliability as does the reciprocating engine, and few cases are on record of a vessel having to be towed into port owing to complete failure of the main turbines. This is important when considering future turbine developments.

In conclusion, the writer adheres to the opinion expressed at the beginning of this comment, namely, that the future of steam will depend principally on the steam turbine and not on the reciprocating steam engine, this also applying to the auxiliary machinery units, such as pumps, electrical generators, etc. Viewed from both a mechanical and thermal basis, it must be admitted that the turbine constitutes an ideal and almost frictionless machine for the "steam-gap" transmission of power, and it is applicable to every type of ocean-going vessel.

Steam for Large Ships Only.

The Superintendent Engineer of the Lancashire Shipping Co., Ltd., expresses interesting views on Diesel and steam propulsion, and strongly recommends mechanical stoking.

By A. E. Ash.

It is common knowledge that due mainly to the inroads into sea propulsion by the internal-combustion engine, both the steam reciprocator and turbine engines have had their fuel consumptions reduced by 30 to 40 per cent in the last four or five years. This has been brought about by high steam pressures, improved boiler efficiency, high superheat, increase in feed temperatures and other minor refinements. Also in the case of the reciprocator, by improved

valves and their actuating gear, and again at the zero end of the "heat range" by the fitting of a turbine and so obtaining useful work at the low-temperature vacuum end. Even so, except for the difference in fuel cost between coal (and its handling) and fuel oils, the steamship with powers below 15,000 horsepower would rapidly disappear, in spite of the higher cost of the Diesel-engined vessel.

You cannot employ a middleman without someone paying, and whereas the Diesel engine converts heat into work direct in the cylinder, the steam unit must employ the middleman and pay through heat losses in the boiler, steam pipes, and engine. The handicap is fundamental and cannot be circumvented.

This handicap would be heavier, but for the fact that the steam unit can take work out of the low end of the "heat range" below the present capacity of the Diesel engine.

Paradoxically, we are now, in Diesel installations, employing the middleman by generating steam in waste-heat boilers and getting useful work from a lower level in the heat scale, and thereby improving the overall thermal efficiency. It is even conceivable that some of the remaining heat left in the gases, after passing through the boilers, could be employed in raising the temperature of the oil fed to the Diesel engine, analogous to the hot feed to the steam boiler. It sounds like a dog feeding on its own tail!

Would the heat of combustion be increased? Microscopic, I agree, but I think one effect would be that the usual compression of air to around 450 pounds per square inch to give a good firing temperature, might be lowered a little, and obviously less work would be used in the compression strokes, *i.e.*, a reduced pumping loss would result.

With a hot oil charge, ignition would be easier and combustion hastened and the maximum and mean pressures probably maintained in spite of lower initial air compression. I wonder!—and remember that we heat the oil we burn under boilers. If the suggestion holds, it means we again get useful work at a still lower level in the heat scale.

Consider our old friend the Scotch boiler—it has many virtues and many defects. It does not lend itself to pressures much beyond 250 pounds per square inch or superheated steam temperatures much above 600 degrees F. without introducing complications which one does not care to face. It has already given place to water-tube boilers for large powers, and for smaller installations, say up to 10,000 horsepower, it may join hands and form a cylindrical-water-tube combination on the lines of the Howden-Johnson boiler. Its efficiency would be much improved if a satisfactory system of mechanical stoking could be evolved, doing away with the continual admission of cold air, while firing by hand and cleaning fires.

FUEL PRICES.

The future of steam propulsion is wrapped up with the difference in price between coal and oils. The day when the Diesel suffered from want of knowledge in design, the lack of suitable materials for cylinders and valves, and its very high cost are past. Its low oil consumption has been mainly translated into increased speed. For long journeys at 14 knots and over, the coal burner is ruled out—the loss of deadweight and the useful cubic space are too serious. Oil is carried in the double bottom, with tonnage as 1 is to 4/5 and cubic as 38 is to 42/28 compared with coal.

For trades around the world where cheap oil in the Pacific is obtainable, also U.K. to Australia run with cheap oil at the half-way house (Panama), it is certain that the steamship will disappear for powers of anything less than 15,000 horsepower.

Turn to Lloyd's Register and study the motor tonnage, turn to the fishing craft, where drifters and even the trawlers are rapidly going over to the oil engine. Again, what do we see going on in the coasting trade? A growing number of oil-driven coasting and cross channel boats; two recently completed are credited with fourteen knots loaded speed.

One factor requires constant consideration and watching, and that is that the world's oil supply is in very few hands, and it is axiomatic that monopolies have no mercy. The oil-fired steamer can return to coal. What of the Diesel? Shall we ever get oil from our coal as a commercial proposition?

Steam and Coal.

A prominent shipowner's views.

By Sir John Latta, Bart.

As a shipowner interested in the fuel question I welcome your efforts to further the cause of steam propulsion. . . . My technical knowledge is not such as to really justify my writing to a journal of the importance of "The Marine Engineer." My line of argument, however, has generally been in regard to the essential necessity of the shipbuilder, engineer and shipowner, marrying themselves together to produce an article calculated to meet the changed conditions which have overtaken the country in the advent of the Diesel engine and the increasing use of oil.

In my judgment, we are all shutting our eyes to an insidious attack which is operating against our very existence. There are innumerable trades that can be conducted just as well by the steam engine, burning coal, as by the Diesel, using foreign oil, the latter simply having the advantage that it creates less worry in its manipulation, but is no more profitable, while its incidence is steadily taking the bread out of our workers' mouths. . . .

The Promise of Recent Boiler Developments.

By J. Hamilton Gibson, Wh.Ex., M.Inst.C.E., M.I.N.A., M.I.Mar.E.

A healthy sign of the times is the renewed interest in high-pressure superheated steam for the propelling machinery of all classes of sea-going vessels. Every rule is said to be proved by its exceptions, and in this case exceptional circumstances can generally be adduced for any departure. But for universal application and adaptability, steam plant remains unrivalled, and successive improvements are slowly but surely bringing down the fuel consumption to figures that compare very favorably with those of heavy-oil engines.

The potentialities of steam power-availing development had been recognized by far-seeing engineers long before that eminent naval architect, the late Sir John Biles, gave the required fillip in a series of remarkable scientific papers, full of unchallengeable facts and figures, culminating in the late Sir Charles Parsons' last great demonstration of high-pressure geared-turbine development in the small Clyde passenger steamship *King George V.* That was but nine years ago, but it was the beginning of a new epoch in steam propulsion, and proved the outstanding advantages of high-pressure steam. It also established once and for all the water-tube boiler in the mercantile marine which, being bound by safety-first considerations, must always proceed with due caution. Within a couple of years, practically all new high-class ocean-going passenger vessels had adopted water-tube boilers, carrying pressures round about 400 pounds per square inch, with a high superheat limited only by the available materials. These and many later big steamships have run for years without incident on regular ocean routes,

maintaining and even improving on their trial trip performances, a feature, in passing, which the advocates of Diesel propulsion claimed as an exclusive feature of the motorship.

Perhaps the most important feature of modern marine machinery is the enormous reduction of weight and of space occupied as compared with only ten years ago, and if present-day developments in super high-pressure steam generation materialize in general adoption, we may look forward to an even more striking saving in space and weight during the next decade.

The large and heavy steam and water drums of existing types of water-tube boilers are eliminated in the new systems of all-tubular super-pressure boilers; and by making use of forced combustion and forced circulation, the steam output per ton of boiler weight is enormously increased. Moreover, the space required is so small that boiler-rooms as such are disappearing; the steam generating plant being accommodated in the engine-room itself. A full head of steam can be raised from cold in five minutes, and sudden calls for more or less steam meet with immediate response by the manipulation of a couple of hand wheels controlling combustion and circulation. Already several such marine installations are in service, and in course of construction. British and foreign naval authorities are keenly interested and are trying out the systems for themselves.

For large powers and high speeds, steam propulsion, whether for geared or electric turbo-transmission, has always been unassailable. That position will be maintained and more firmly established by the advent of higher pressures and superheats. And recent developments in simplified steam turbine practice for the smaller powers and slower ship speeds required in what is known as tramp service, indicate that superheated steam, raised by coal or oil fuel, in any and every type of marine boiler, will continue to play a major part in sea-borne commerce.

New Combination Machinery Proposal.

The designer of the claimed most economical marine steam engine has some interesting views on the further development of combination machinery in association with electric drive.

By Wm. Albert White, M.N.E.C. Inst., M.I.Mar.E.

I visualize three distinct developments in steam propulsion, and these may be summarized as follows:—

(1) For the steam propulsion of the large high-powered liner, steam pressures and superheat temperatures will continue to advance towards the Mecca of highest economy and maximum power obtainable when using the lowest possible weight of machinery. During recent years material advances have been made in this respect. We are now assured of a modern liner's propulsive equipment for a given power being concentrated in approximately half the space, and of having half the weight of her progenitor of but five or six years ago, and using steam pressures and superheats limited only by the strength and cost of the materials of construction used. Feed temperatures approaching the boiler steam temperature will be aimed for.

For this class of vessel the steam turbine will continue to be the prime mover.

(2) The steam propulsion of the cargo-carrying vessel on competitive routes, and of necessity so equipped that she can trade away from home for an unlimited time, sets its own limitation to steam pressures and superheats. The latter are governed largely by the limiting pressure of the multitubular or Scotch boiler, which is still first favorite for the cargo carrier, and looks

like holding this position for some years at least. Now that the power required to drive the cargo carrier at an economical speed is so small, there is no necessity to go to high pressures and superheats. While the gain from high pressures and temperatures is granted, the actual amount saved in fuel, owing to the already low fuel consumption obtaining in terms of tons per day, does not warrant the increased cost and necessary safeguards required. We therefore arrive at pressures of 240 to 260 pounds per square inch as possibly the pressures for the next few years for the cargo carrier, and superheats of 620 degrees to 650 degrees F.

COMBINATION MACHINERY.

As regards type of machinery, for power up to, say, 4000 I.H.P. per engine, the maximum economy from any steam unit will be obtained from the combined high-speed, balanced, force-lubricated reciprocating engine with reheated steam to the low-pressure cylinder, and exhausting through a reheater to the turbine, both engine and turbine transmitting their power to the propeller shaft through helical gearing.

(3) The combination of steam unit with electric drive to the propeller is a development of the future which must be considered. The steam unit would be constructed to operate in one direction only (reversing being done by the electrical apparatus), and could consist of the combination already mentioned with the speed of the enclosed, force-lubricated, reciprocating engine greatly increased. This would result in a material reduction in engine size and weight, thus giving a simple, cheap and efficient unit, requiring the minimum of attendance, and further, one which could be readily operated from the deck.

I anticipate that the future will see great developments along this line, as it only awaits the electrical engineer to give us a reasonably economical and commercial apparatus for transferring the power given by the steam unit on to the propeller shaft in place of the present heavy, cumbersome and expensive arrangements of electric transmission equipment.

New Conceptions of Steam Propulsion.

Best known as a motorship protagonist, the author of this comment here expresses confidence in the latest boiler advances.

By A. C. Hardy.

Le roi est mort . . . vive le roi!—This well-worn tag is not at this juncture inappropriate to the situation as regards steam propelling machinery. The old "Roi" is undoubtedly dead, with its low-pressure, unsuperheated boilers, and its inefficient machines, both mechanically and thermo-dynamically, but the new "Roi" is so very much alive that it is likely to have quite a modifying effect upon the philosophy of marine propulsion generally. The battle, it would seem, in this connection will not be one between coal and oil, for although we cannot by any means rule out coal entirely, its uses will be severely restricted to special purposes in the future, and, in any case, the consumption per horsepower hour will be much lower than it has been before. Rather will it concern internal and external combustion—oil in any case: coal dust perhaps.

NEW STEAM GENERATORS.

Steam of the future is steam by courtesy title only, because it is likely to be produced in high-pressure generators, and the term boiler is no longer suitable. The type of generator in fact, is one which can be raised from

cold to a full head in a few minutes, and which will be so compact that it can be placed alongside the prime mover to which it supplies power. Thus we shall have the flexibility and startability of the Diesel engine, if not the extremely low fuel consumption of the latter. This will induce further effort on the part of the Diesel engine manufacturers to reduce their fuel consumption and decrease the space required for a given power, so that it will be seen that this development is likely in the future to take us to unheard-of economies in the way of weight and space.

It is a known fact that the Diesel engine is developing to a point where practically 3000 B.H.P. per double-acting cylinder is a workable proposition today, and it requires but little imagination to see what power this means in a cargo liner ship of twin screws, with two three-cylinder engines of this type. It is in cases where the double-acting two-stroke engine is employable that even this modern fleet will have its greatest difficulties to compete, while in certain specific cases, such as coasters and the like and small passenger ships, the Diesel engine is likely to reign supreme for some time to come.

DIESEL BUILDERS AND STEAM DEVELOPMENTS.

In the passenger-carrying category, whether it be large liner or cross-Channel vessel, the steam generator of one or another of the type put forward at the recent Institute of Marine Engineers' symposium, is likely to have a far-reaching effect. Possibly, we shall learn much from naval practice in this respect. In any case this development has now gone too far to be considered merely a laboratory experiment. It is a significant fact that some of the most prominent manufacturers of Diesel engines are among the foremost in developing these new high-duty, light-weight steam generators. This indicates a rapprochement between steam and Diesel, and shows rather clearly that the line of development at the present moment appears to lie in the direction of making external combustion approach internal combustion. At all events the situation is as pregnant with interest as it ever was.

Cargo Ship Simplicity.

The superintendent engineer of the Prince Line holds conservative views.

By S. N. Kent.

The interested observer is possibly looking forward with keen anticipation to the future developments of marine propulsion and seeking to learn if steam will find favor again for cargo liners, tankers, and those other types of vessel where Diesel propulsion is at present most popular. For steamships there are two major aspects of the propulsion problem: oil or coal. It is absurd to imagine that oil can be ousted from naval vessels, although several naval men have recently written to the daily Press regarding the desirability of our having either coal-fired warships or ones capable of rapid fuel change-over. For large fast passenger vessels, also, coal is today unthinkable. For the tramp and the smaller and important fast cargo liners, the question of propulsion resolves itself into steam or internal combustion. For long voyages, and where oil for burning under boilers is obtainable at cheap rates, the steam engine is well worthy of consideration. To carry oil bunkers, settling tanks, oil separators, double-bottom tanks specially constructed for oil carrying, necessitated these things being paid for; and in the case of repairs to double-bottom tanks for carrying oil, extra charges are

made for internal inspection, repairs, damage, etc. Slack tanks cannot be avoided. Pipe lines for oil are a necessity, and extra pumps, etc., must be provided. Every item means staff, overhauling expenses, spaces, repairs, jointings, etc. On the credit side, however, there are savings in time in loading, trimming and transferring fuel and in keeping the machinery spaces clean. The case for coal is too well known to call for comment on this occasion.

The triple-expansion engine, reasonably superheated, with a Bauer-Wach, Gotaverken and/or White additions, and designed for limited pressures on all bearings, a good vacuum based on a cooling-water temperature of 80-86 degrees F., and having as many as possible of the pumps driven by links off the engine, should and does make possible a very economical installation. Turning to boilers, here we are faced with a problem. The many types of high-pressure boiler recently introduced, as discussed and illustrated at the Institute of Marine Engineers on March 12 last, seems to indicate that competition among makers of these new types is causing them to avoid each other's patents and each to go one better than his neighbor. Were a united effort instituted and a single design evolved this very important new development might find salvation in something really practical and commercially efficient. As it is at present, I think too much effort is likely to be dissipated on unnecessary competition before the business really gets moving. The old multitubular boiler will give good service for years without repairs, given reasonable care and attention, in good weather and bad, and if a tube fails it does not take long to remedy. The boiler can also be examined very thoroughly when opened up, and so far as experience goes these boilers—without appendages—have proved to give the best service in the long run. Can the same be said of alternative types? It is well to ask, in fact, whether *very* high pressures and temperatures are worth while, despite the economy of these boilers, for personally, speaking as a practical user of boilers, I doubt it.

For home trading I still favor a good steam-driven tramp with three adequate Scotch boilers having firebars 4 feet 0 inch long (for considerations of manhandling), reasonable superheat, tubular air heaters, small pressure-forced draught, super feed-heaters, and as many pumps as possible driven by the links (which should have roller bearings). For the main triple engine I favor poppet valves for the H.P. and balanced slides for the other cylinders. A turbine can be introduced either between stages of expansion or after the L.P. cylinder, and to the turbine I would append a rotary feed pump, and belt or chain drive for a fan and/or lighting dynamo. Alternatively we could have a larger dynamo, for driving the motors for certain engine-room pumps, domestic refrigeration plant, as well as for lighting. Thus all the machinery would be "triple-expansion." Would this not give us 1 pound coal per I.H.P. per hour for all purposes, a job which all engineers understand, moderate upkeep, simplicity, a good engine-room layout and, 12 years hence, still a going concern?

Auxiliary Propeller Drive.

An interesting combination machinery development is put forward.

By E. Carlton Garratt.

A few years ago it seemed that steam engines for marine purposes were to be eclipsed by the internal-combustion engine. Since then the position has altered considerably, and it is evident that steam has regained much of the favor which it lost in the first enthusiasm for Diesels.

I am considering here chiefly the cargo vessel or tanker of from 2000 to 4000 I.H.P., and which is usually running on tramp service. Where these vessels are fitted with steam reciprocating engines, much progress has been made in improving their efficiency by the fitting of a low-pressure exhaust turbine, and I think that it is in this direction that we shall see further progress in the near future.

At first glance it seems only logical that the turbine with its high efficiency and small dimensions should be increasingly used, and within the last few months very interesting designs for an all-turbine unit of about 2000 I.H.P. have been given a good deal of publicity, yet the past history of high-pressure turbines for vessels of the tramp class has not been encouraging. The usual conditions of operation are particularly unsuitable for the care and accuracy essential for the successful operation of high-pressure turbines, and it is an accepted fact that the high-pressure turbines have not been doing their share of the work for which they were fitted, but experience has shown that exhaust turbines, coupled to reciprocating engines, are being run under very exacting conditions quite successfully, and I think that this type of engine will come increasingly into favor.

There are numerous systems employing an exhaust turbine installation to be considered. I have discussed this question elsewhere, and so will only repeat here that, for thermo-dynamical reasons, the only systems which can take full advantage of the extra power developed by the turbine are those which use it for mechanical power, not those which put it back into the main engine entirely or partly in the form of heat. This rules out the majority, and leaves such systems as Bauer-Wach, White's combination machinery, and the auxiliary propeller drive. As I am personally interested in this latter (my father, Mr. E. A. Garratt, is the patentee of the system), it is difficult to be unbiased, but I think it can fairly be said that it has very definite advantages. It is simple, highly efficient, and very considerably cheaper than the usual form of exhaust-steam turbine plant. The self-propelled model tests carried out in the experiment tank at the National Physical Laboratory established that the auxiliary propeller in no way affects the capability of a vessel to steam in a straight line without use of helm. So, for the tramp class of vessel, I expect to see an increasing proportion of steam engines of the reciprocating type using high-pressure superheated steam and coupled to an exhaust turbine, the turbine being either geared back to the main shaft or driving an auxiliary propeller, either through single-reduction gearing or electrically by means of a turbo-generator and an A.C. motor mounted on the auxiliary shaft. The direct system is suited for aft-end engine-rooms and the electrical system either for amidships or aft-end engine-rooms.

Striking Comparative Figures.

The Managing Director of the Superheater Co., Ltd., quotes some interesting performance data.

By Eric A. Robinson.

Progress in marine steam propulsion within the last half-century is well illustrated by comparing the results of the Cunarder *Servia*, built in 1881, with the Canadian Pacific line *Empress of Britain* fifty years later. The *Servia* (90 pounds per square inch boiler pressure) consumed about 2.8 pounds of coal per equivalent S.H.P. per hour delivered to the propeller, or about 38,000 B.T.U.'s per S.H.P. per hour.

The *Empress of Britain* (425 pounds per square inch boiler pressure) has a fuel oil consumption, all purposes, of 0.57 pound per S.H.P. per hour or an

equivalent coal consumption of 0.775 pound, *i.e.*, 11,000 B.T.U.'s per S.H.P. per hour. That is, the overall efficiency from bunker to propeller has increased in the ratio of 7 per cent to 23 per cent, 2 B.T.U.'s doing the same work as 7 B.T.U.'s fifty years ago. These figures are not, however, strictly comparable, owing to Diesel auxiliaries in the latter vessel, but they are correct as regards relative overall efficiencies.

As 70 per cent of the total heat in the steam, or about 63 per cent of the total heat in the fuel, even with the most modern boilers and turbines, is lost in the condenser, further improvements in fuel economy can only be relatively small considering the commercial factors of first cost and maintenance.

On the steam generating side, the primary features for increased economy are higher steam temperature and waste-heat regeneration. For the ranges between 1000 and up to 4000 I.H.P. per ship, embracing approximately 90 per cent of the world's tonnage, a boiler pressure of 300 pounds per square inch with superheaters to give a total temperature of 750 degrees F. may be recommended to obtain the best practical results, taking operating conditions into consideration. For this class of ship, in which Scotch boilers are usually fitted, an improved cylindrical boiler, such as the Howden-Johnson, with a rotating air preheater, merits careful consideration. This type of boiler enables practically the whole of the heat to be recovered from the gases, so that an overall boiler efficiency of 90 per cent can be obtained with service evaporation of 8 pounds per square foot of boiler heating surface.

Reciprocating engines, with H.P. poppet valves of the Lentz type, preferably double-compound, with admission steam temperature of 700 degrees F. and exhaust turbine with high vacuum, will give an economy of over 40 per cent as compared with ordinary engines using saturated steam. Adding the improved efficiency of steam generators, a total economy of 55 per cent will be obtained, or a consumption of 0.9 pound of coal per I.H.P. per hour (1 pound per equivalent S.H.P.) or 0.7 pound fuel oil per S.H.P. per hour. A large proportion of our freight is, however, today still carried in steamships where the coal consumption averages as high as 1.6 pounds per I.H.P. per hour.

The results obtained from the use of boilers and engines of the latest designs compare favorably, therefore, with Diesel engines, having a consumption of 0.35 pound of oil per S.H.P. per hour when consideration is given to the higher cost of Diesel fuel oil and lubricating oil, together with increased capital outlay and maintenance.—"The Marine Engineer," April, 1935.

BURNING OF CRACKED-RESIDUE FUEL OILS.*

By M. J. Hanlon.

Black Diamond Steamship Corporation.

Because of increasing fuel-oil prices, it is necessary that those charged with responsibility for economical and efficient power generation, afloat and ashore, pay particular attention to the methods being employed in the combustion of fuel oil of any type—cracked residue, bunker "C," or high grade.

A tremendous wastage, with attendant uneconomical results, may occur through inattention to the proper functioning of oil heaters, strainers, and burners. Particular attention must also be given to the admission of air; and the quantity, velocity, and temperature of the air admitted to the furnace

* Presented at a meeting in New York, N. Y., Nov. 8, 1934, of the Metropolitan Section of "The American Society of Mechanical Engineers."

are important factors in securing the desired results. The pressures and temperatures maintained on the fuel oil are also of extreme importance in order that the fuel may be atomized as finely as possible with elimination of streaks and globules so that the most intimate mixture of air and fuel may be secured and the maximum combustion maintained at all times.

The quantity of air admitted to the furnace per pound of fuel oil being consumed must be closely regulated. It has been determined in marine practice that about 16 pounds of air per pound of fuel oil gives the best results, and any admission of air much in excess of this amount simply means that fuel is being wasted in heating the unnecessary air admitted to the furnace.

The burner tips or atomizers may become badly carbonized or coated with a coke-like substance caused by the burning back of the flame on to the tip. This can be easily remedied by adjusting the burner further back from the furnace.

It is also advantageous to increase air turbulence, and to permit access of additional air to the root of the flame immediately at the burner tip. Special flame cones with angular slots to produce turbulence and circumferential slots to provide flame-root air have been developed and successfully used.

It will be found that the pressures and temperatures necessary to assure proper atomization and efficient combustion will vary over a wide range and will be controlled in a large measure by the type of burner equipment and by fuel characteristics. Some types of burner equipment will require comparatively low pressures, and others will demand a much higher pressure.

No hard and fast rule can be laid down for burner and air regulation and adjustment. The problem finally will resolve itself into a process of experimentation until the best conditions are obtained. Slight changes in the adjustment of burners and air registers, as well as pressures and temperatures, will be found to produce remarkable results.

The time required on the part of the operators to arrive at a proper determination of adjustments will be justified by the results secured.

During the last few years the gasoline demand has been reflected in the fuel-oil market by a fuel type entirely different from that previously available. It made its appearance in the marine field in 1931, and immediately was responsible for considerable difficulties. Operators were not familiar with it, and the fuel was not at first uniform in its characteristics. The burning equipment with which steamers were fitted was originally designed for handling the older standard fuel known as bunker "C"—about 16 A.P.I. gravity, the specific gravity of which is 0.9593.

CRACKED-RESIDUE FUEL OILS.

Although prior to 1932 the A.P.I. gravity of the then available oils dropped to about 10, it was still possible to cope successfully with this fuel without making extensive changes in the existing equipment. With the advent of this new cracked residue on the market, at an attractive price, and the general understanding that it would soon be the only fuel available, it was obvious that oil-burning equipment on steamers would have to be adapted to deal successfully with this fuel, also it was clear that a program of educating operators would have to be undertaken, and many of the older theories and practices in connection with fuel-oil burnings discarded.

A few of the first difficulties experienced with this new type of oil were the plugging of heaters, strainers, and burner tips, and, not infrequently, the bursting of the strainer bodies and strainer baskets, all due to the deposition of flocculent carbon under the influence of elevated temperatures. This is one characteristic of cracked-residue fuel oils, and one of the most difficult to contend with. There is also a decided tendency to mass carbon formation

in the furnace which results in flame impingement on the mass of carbon. If not attended to, this is manifested in poor combustion and the production of smoke.

The specific gravity of cracked-residue fuel oils is much greater than that of oils formerly obtainable, but the viscosity is much lower. During a period of 18 years the fuel oil available for power purposes has shown a marked increase in weight of from 7.73 pounds per gallon in 1916, to 8.6 and 8.8 pounds per gallon in 1934.

As the viscosity of the new cracked-residue fuel oils is much lower than that of the old standard bunker "C" oils, it is unnecessary to employ a temperature as great as that for the higher grade fuels to insure proper atomization. This means lower steam consumption for the heaters, which is an important item from an economy standpoint.

The atomization needed to give the best results can be secured if the oil is preheated at delivery to the burners so that the viscosity is between 3 and 5 degrees Engler or, say, an average of 4 degrees Engler. The corresponding Saybolt universal viscosimeter readings for 3, 5, and 4 Engler will be 105, 181, and 143 seconds, respectively.

Some cracked residues have gravity characteristics of the order of 10 degrees A.P.I. and a viscosity characteristic so low that they will require preheating to only 125 to 150 F. in order to insure proper atomization. As a point of comparison 10 A.P.I. gravity uncracked oil would be heated to approximately 300 F. for proper atomization and efficient combustion.

In the case of uncracked fuel oils the viscosity and gravity had such an interrelation when their characteristics were expressed in degrees A.P.I. that the prediction of correct preheating could be based on the A.P.I. or Baumé gravity. This does not hold true when dealing with the cracked residues.

The cracked residues should be preheated only on the *basis of viscosity* if proper atomization is to be secured with attendant efficient combustion.

When the temperature elevation is excessive in the case of cracked-residue fuels there is a pronounced tendency to flocculent carbon deposits in all high-temperature elements of the oil-burning equipment. These carbon deposits closely resemble what is known as petroleum coke and appear in the furnace as a shower of sparks. They are discharged into the furnace through the atomizer or burner tip as solid particles of carbon or petroleum coke. If this condition exists there is a strong likelihood of forming ash particles which will eventually adhere to the furnace refractory linings and result in heavy slagging and expensive repairs.

In the combustion of uncracked fuel oils this ash apparently does not exist to any appreciable degree, or it may be absorbed in the gas stream and passed out through the stack in infinitesimal particles, although small quantities may remain in the furnace and adhere to the boiler tubes. When burning cracked-residue fuel oils a greater proportion of this ash remains in the furnace with resultant furnace slagging. A considerable quantity of ash and unconsumed carbon may also be deposited in the furnace or carried up the stack.

ESSENTIALS FOR EFFICIENT COMBUSTION.

It is essential to the efficient combustion of these cracked-residue fuel oils that the following points be closely observed:

(1) Provide correct preheating based on the viscosity of the fuel oil.
(2) Maintain a high furnace temperature.
(3) Adjust for long flame travel.
(4) Complete flame propagation in two-thirds the length of the furnace.

(5) Adjust air regulation to provide the requisite quantity by weight per pound of fuel oil being consumed.
(6) Keep excess air admitted to the furnace at a minimum, as it is a waste of fuel to heat the excess air. Fuel is the sole source of heat and costs money.
(7) As an additional precaution against slagging, treat the furnace refractory linings frequently with a coating of high-temperature cement.
(8) Maintain fireside surfaces of boiler tubes in a clean condition.
(9) Guard against outside air leaks through boiler casings.
(10) Check stack temperatures.
(11) Analyze flue gases as a check against excess air, and to correct air regulation.
(12) Maintain oil preheaters, air heaters, suction strainers, discharge strainers, fuel-oil lines, burner tubes, and atomizers in a clean and efficient condition. Provide air turbulence.
(13) Do not allow furnace carbon to build up in mass formation.
(14) Do not allow flame impingement on carbon formations of furnace linings—this will result in the production of smoke and inefficient combustion.

An authority* on the manufacture of this fuel oil says that in treating of fuel oils having their origin in the Gulf Coast area the refinery processes consist of two major operations:

"(*a*) The crude oil is distilled or topped, yielding gasoline with minor by-products such as gas oil, and a fuel-oil residue.
"(*b*) This residue may be subsequently cracked to produce gasoline and a cracked-fuel oil residuum."

It will be observed that here are two separate and distinct operations or processes, the first of which produces a *residue* and the second produces what is termed *residuum*.

The same authority states that the method of operation of cracking processes "may be divided into two classes":

(1) Cracking to low level or liquid residuum, and
(2) Non-residuum operation, *i.e.*, cracking to solid residue of coke.

By using method (1) there is produced about 41 per cent gasoline, 12 per cent gas oil, and 36 per cent fuel oil. Then we are told that by using method (2), or non-residuum, the gasoline yield is increased to 49 per cent, gas oil 14 per cent, and the fuel oil is reduced to 7 per cent, with a yield of about 70 pounds of coke per barrel input. The method selected for operation depends on several factors, such as gasoline price, coke price, fuel oil price, type of cracking unit used, etc.

It is further stated

that by means of the above-described topping or cracking operations four different types of fuel oils may be produced as follows:

(*A*) A straight reduced crude, having a high viscosity and usually a high pour point.
(*B*) A fuel oil composed entirely of cracked residuum, having a low viscosity for a given gravity and a low pour point.
(*C*) A blend of topped crude with a high viscosity and cracked residuum with a low viscosity to give a medium-viscosity fuel oil.
(*D*) A cracked residuum run to a gravity of 10 degrees A.P.I. or lower, then blended with an overhead distillate (preferably a cracked distillate) to a gravity of about 12 degrees A.P.I. This results in a very low-viscosity fuel oil.

* R. T. Goodwin, Shell Petroleum Corporation.

It would seem that the reduced-crude fuel oils described under (*A*) have been supplied in the past because of the extremely low market price for crude oils and limited cracking facilities available. The reduced-crude fuel oils rapidly disappear from the market as soon as the prices of crudes and consequently the price of gasoline are increased. These high prices lead to the production of a fuel oil described under (*B*), which is composed entirely of cracked residuum.

Residue or residuum at 10 degrees A.P.I. are acceptable and could be satisfactorily dealt with, and at 10 degrees A.P.I. would have the specific gravity of water. But some of the cracked residues have an A.P.I. gravity as low as 2.0 degrees and a specific gravity of 1.06. Some of the characteristics of this fuel will be given later.

Cracked-residue fuels are excellent fuels for power purposes provided the storage tanks in power plants, or the double-bottom tanks, deep tanks, and settling tanks in steamers, as well as the fuel-oil-burning equipment afloat and ashore, are adapted for the use of this fuel.

CHARACTERISTICS OF CRACKED-RESIDUE OILS.

The cracked residues possess the following desirable characteristics:

(1) They have a low viscosity for a high specific gravity.

(2) A rapid break occurs in the viscosity of the fuel with temperature increase usually at from 130 to 155 F.

(3) The pour point varies from 0 to 15 F. This feature is most important in marine practice where extremely cold water is encountered in winter.

(4) The heating values of uncracked and cracked-residue fuel are about equal. The cracked residue is usually lower in gravity and weighs more and results in more pounds of fuel oil per gallon or barrel.

The former U. S. Navy standard specifications for bunker "C" fuel have been superseded by the following:

(*A*) Navy fuel oil shall be a hydrocarbon oil, free from grit, acid, and fibrous or other foreign matter likely to clog or injure the burners or valves.

(*B*) The flash point shall not be lower than 150 F. when tested in a Pensky-Martens closed tester.

(*C*) The viscosity shall be measured at two points as follows:

(1) The viscosity in Saybolt Furol seconds shall be not greater than 750 seconds at 77 F.

(2) The temperature at which the viscosity in Saybolt universal seconds is 150 shall be not greater than 185 F.

The oil as delivered shall not contain more than 1.0 per cent of total water. For deliveries by supplier's tankers or barge, oil containing not more than 2.0 per cent of total water is acceptable with a reduction in price for all water in excess of 1.0 per cent.

NOTE: The reduction shall be (1.25 times excess water per cent) per cent of contract price.

(*D*) Ash shall not be greater than 0.12 per cent.

(*E*) The oil shall have a fire point of not less than 220 F.

If the water content of the oil prevents an accurate determination of fire point, the oil will be acceptable if the foaming due to water starts below the fire point and at a temperature not lower than 195 F. If the flash point is 175 F. or higher, the fire point need not be determined.

(*F*) The oil shall be free from a tendency to deposit excessively in fuel-oil heaters when being heated for atomization at a viscosity of 150 S.S.U., for

all rates of flow from at anchor to full power, with sudden changes of rate of flow and with steam to oil heater manually controlled.

(*G*) The oil shall be free from a tendency to sludge excessively in storage at temperatures from 40 F. to 100 F.

In view of these specifications Table 1 will be of interest as it gives some idea of the specifications offered by certain oil refiners and on which large quantities of a cracked-residue (or residuum) fuel oil were purchased and was and is today efficiently consumed in marine practice. Particular attention is directed to the A.P.I. gravity of this fuel. The analyses given were not made by the oil refiners, but were made for the buyer by an independent laboratory of the highest standing.

TABLE 1—CONTRACT SPECIFICATIONS.

Btu. per lb. (17500)	Viscosity, Saybolt-Furol at 122 F. 125 max.	Actual Analysis			Flash 150 min., F.	A.P.I. 2 to 5 deg. range	Pour pt. 30 max., F.
		Sediment 0.25 max., per cent	B.S.&W. 1 max., per cent	Sulphur 2.0 max., per cent			
17,778	129	0.63	0.4	1.05	202	2.4	10
17,859	127	0.30	0.3	0.89	200	2.6	10
17,753	117	0.23	0.6	0.86	194	3.2	15
17,760	123	0.21	0.4	0.89	198	3.3	10
17,792	132	0.24	0.5	0.94	204	3.0	15
17,864	93	0.15	0.4	0.92	200	2.6	10
17,804	102	0.37	0.5	0.89	190	2.7	10
17,825	98	0.34	0.2	0.89	194	2.6	10
17,944	95	0.24	0.4	0.90	200	3.7	15

It will be noted that the viscosity is low—very low in some of the analyses—and the specific gravity expressed in terms of A.P.I. gravity is very high.

Some figures on the specific-gravity characteristics of this fuel may also be of interest. The A.P.I. gravity range specified is from 2 to 5 degrees. Now 2 A.P.I. gravity fuel oil has a specific gravity of 1.06, so that it is heavier than and will sink in fresh water. It weighs 370.8 pounds per barrel of 42 U. S. gallons. There are 6.041 barrels per ton.

The 5 A.P.I. gravity fuel oil has a specific gravity of 1.036, it is also heavier than fresh water. It weighs 362.6 pounds per barrel of 42 U. S. gallons, and a ton will contain 6.18 barrels. The heating value specified is 17,500 B.T.U. per pound and analysis showed a better figure. The average heating value for the fuel in the 2 to 5 A.P.I. gravity range is 6,417,250 B.T.U. per barrel, and 39,200,000 B.T.U. per long ton.

The analysis approaching nearest to the original bunker "C" available is that of a 14 A.P.I. gravity fuel which has a specific gravity of 0.9725. It is lighter than fresh water. It weighs 340.158 pounds per U. S. gallon, and a ton will contain 6.496 barrels. The average heating value is 18,500 B.T.U. per pound or 6,292,923 B.T.U. per barrel and 41,400,000 B.T.U. per long ton.

A third type of high-grade fuel included in this comparison has an A.P.I. gravity range of from 19 to about 27.45 degrees. The specific gravity ranges from 0.940 to 0.890. For the high-grade oil at a gravity midway between 0.890 and 0.940, say, about 0.915, equal to about 23.14 A.P.I. there are 294.049 U. S. gallons, or 7.001 U. S. barrels of 42 gallons per long ton. This oil weighs 319.914 pounds per barrel. The heating value is 5,998,387 B.T.U. per barrel and 42,000,000 B.T.U. per long ton.

COMPARISONS OF THREE OILS.

A summary of the heating values of these three types of fuel oil follows:

B.T.U. content per ton of 2240 pounds	
(1) High-grade fuel	42,000,000
(2) Approximate bunker "C"	41,400,000
(3) Cracked residue	39,200,000
B.T.U. content per barrel of 42 U. S. gallons	
(1) Cracked residue	6,417,250
(2) Approximate bunker "C"	6,292,923
(3) High-grade fuel	5,998,387

The standard unit of purchase in fuel-oil contracts for marine purposes in the United States is the barrel of 42 U. S. gallons. It is apparent, therefore, that the cracked-residue oils, due to their higher specific gravity, yield more heat units per barrel and per gallon than either the so-called bunker "C" or higher grade fuels.

In referring to atomization of fuel oils, mechanical atomization by pressure is implied.

The type of fuel preheater that has given best results with cracked-residue oils is that provided with a narrow annular space of large sectional area through which the fuel moves spirally at a low velocity. Steam at a comparatively low pressure is used for heating and the possibility of "shock heating" the fuel is obviated. When necessary this type of heater is easily opened up and cleaned.

The atomizer proper, or burner tip, should be of the type that has a high centrifugal characteristic to secure the finest possible atomization. The flame characteristic must be such that there will be no impingement on the back wall, side wall, or furnace floors. Also, the spacing of the burner units must be such that there will be no overlapping of flame-zone areas.

Preheating the air is of great assistance in successfully burning the cracked-residue oils, aside from its efficiency aspect as a means of heat recovery.

If the procedure that has been outlined is followed, and the detail changes and adjustments are made which experience in each individual power plant will indicate, the new type of cracked-residue fuel oils can be successfully and economically burned.—"Mechanical Engineering," April, 1935.

"WHY THE UNITED STATES REQUIRES A MERCHANT MARINE."

In the Unanimous Opinion of the Judges in the Propeller Club Nation-Wide Essay Contest, This Was Considered the Best Essay Submitted by Any of the Contestants.

By NORMAN O. W. ADAMS, JR.

The pages of history teach us in a most convincing manner that those nations which have attained commercial supremacy and achieved national security have done so by fostering a Merchant Marine and by protecting their trade routes, their merchants, and their merchant ships against all aggression.

The role of the Merchant Marine in our national economic development is not a recent discovery, nor has its growth been sudden. Its early history is resplendent with the exploits of the clipper ships whose house-flags were whipped by every breeze of the Seven Seas. These American vehicles of commerce were noted for their speed, efficiency, and modernity and comprised

a Merchant Marine that ranked second only to that of Great Britain. Carrying as high as 77 per cent of our products, this fleet of trim sailing vessels was a prime factor in establishing our national security and well-being.

Further progress in the development of this national asset was halted by the sudden shift of emphasis in 1861 with the advent of the Civil War. The introduction of iron and steel ships by Great Britain, replacing our vessels of wood and sail, shortly thereafter contributed to the decline and virtual disappearance of a merchant fleet that had once been the pride of these United States. As a matter of fact, our Merchant Marine might have disappeared entirely had we not sustained a formidable shock to our increasing indifference and lethargy by the impact of the World War in 1914. Then and then only did we recognize that our national well-being and economic independence necessitated sea power of the first order and an adequate Merchant Marine.

As to the wartime value of the Merchant Marine, it has been truly said that a navy without such an accessory is a navy in name only. How could our navy be expected to conduct a major campaign unless supported by a sufficient number of auxiliaries obtained from our Merchant Fleet? In this group of auxiliary ships are to be found ammunition ships, cable ships, colliers, tankers, refrigerator ships, supply transports, auxiliary cruisers and other craft that would be normally requisitioned in time of national crises. Our total inadequacy of ships of the type mentioned was graphically demonstrated in the recent world conflict when more than eight months were required to transport an army of 2,000,000 American soldiers to the battlefields of Europe. Every patriotic American would do well to consider the following table compiled by the United States War Department, which gives the total number of American soldiers and war personnel carried in trans-Atlantic travel to France: By American ships, 987,726; British, 1,114,587; French, 47,701; other ships, 4698—a total of 2,154,712. Thus, it is significant to note that more American soldiers were carried to war on ships flying the British colors than on vessels of all other flags, including our own.

With war clouds hovering over Europe and Asia at the present time, it is imperative that we inventory our auxiliary fleet to determine our preparedness in case of another war. Likewise, worthy of recognition is the fact that as navies are reduced, the Merchant Marine of a country becomes its navy in exact proportion. Sink today all the naval vessels in the world and the nation having the largest Merchant Marine is instantly the possessor of the world's largest navy. War, like history, repeats itself, and these United States can ill afford to face the possibility of another major conflict with an inadequate Merchant Marine such as we had in 1918. Rear Admiral E. B. Fenner has presented the navy's attitude toward a Merchant Marine in these words: "A Powerful Navy and a Merchant Marine go hand in hand to make a great nation, and without them a nation will not be great for long."

In 1914 conditions brought about by the World War caused foreign ships, upon which we had depended for the transporting of our goods, to be withdrawn from our trade. We then relied upon Great Britain, France, Germany, Italy, Austria and other nations for the tonnage which we lacked. Then, with the advent of this war, their ships were withdrawn just as Washington, Jefferson, Madison, and other illustrious witnesses predicted that they would be. Cargoes destined for the nations of the world congested our yards, docks, terminals, and railroad sidings for miles. The North, South, East and West, all alike, felt the destructive force of our misguided policies. To cite an authentic condition: Before the United States entered the war, Europe begged American cotton growers to sell them cotton. The demand was great and urgent, but American exporters were forced to inform their potential purchasers that we had ample stocks of cotton available, but no ships of our own in which to complete delivery. European ships were needed

for war purposes and to carry troops and supplies at this crucial period. The direct result was that American cotton growers lost millions because deliveries could not be made, while warehouses were overflowing and hundreds of bales of cotton had to be literally stacked in the streets and public squares.

Thus, fascinated by the lure of easy living and succumbing to the arguments of expediency, we actually tried the experiment which we had been warned against by every leading statesman since the foundation of the Republic. In this manner we paid the inevitable price of our folly in 1914, when the mad scramble for tonnage showed how completely we had abdicated the proud position won on the Seven Seas by the Yankee Clippers. This unpardonable predicament forced us to embark on a superhuman program of ship building that cost us upward of $3,000,000,000.00. Had that fabulous sum been leisurely and wisely and systematically applied during the previous generation to creating an all-sufficient American Merchant Marine, we should not have suffered the humiliation we then experienced, nor have had to add practically that sum to the other enormous war debts.

From its earliest history as a nation America has found her destiny on the sea by reason of her natural and geographical make-up. No nation possesses such an extensive coast line as do these United States. We have 7000 miles of continental shelf and an equal frontage on the Great Lakes, in addition to 15,000 miles of navigable rivers. Only Russia has frontage on both the Atlantic and Pacific Oceans and the harbors of this nation are ice-locked in the West and practically cut off by alien territory in the East. No nation is so ideally situated to take advantage of ocean transportation as is our own country.

In this period of economic depression we should consider a very remarkable coincidence: There has been prosperity in the United States only during those times when foreign trade was flourishing. It is equally significant that employment has reached its highest level when foreign trade was greatest. Therefore, it behooves us to encourage, foster, and maintain foreign trade, which is of utmost importance in assuring the return of prosperity and our national well-being. Assuming that a revival of prosperity is contingent upon the rehabilitation of our foreign trade, our activities to this end are of twofold nature. Foreign markets must be sought after and American merchant ships must be provided for the export of those products that emanate from our farms and factories.

A growing demand in foreign markets for such American products as cotton, nickel, farm equipment, automobiles, and glass is convincing proof of the first of these foregoing principles. During 1934, one-sixth of American farm products was sent abroad. In one year we exported 54 per cent of our cotton, 41 per cent of our tobacco, 40 per cent of our typewriters, 36 per cent of our kerosene, 33 per cent of our lard and 25 per cent of our sewing machines. There is hardly a town in this country that does not produce something for export. America has the raw materials and the finished and semi-finished products that the world wants and if provided with a modern and efficient American Merchant Marine could regain her former position of commercial ascendency.

Not only is the foregoing a review of actual facts, but also, a matter of vital importance because only 23 per cent of American labor is required to feed, clothe, and shelter our entire nation, while the other 77 per cent of our employable force must depend to a great extent on exports and their medium of transportation, the American Merchant Marine, for a livelihood.

Our present Merchant Marine, though inadequate, and to a great extent obsolete, gives employment to some 200,000 men. Other thousands find work on harbor craft, on the piers, and in the offices of steamship companies.

Shipbuilding, an industry intimately associated with the Merchant Marine, gives employment to from 20,000 to 25,000 persons in shipyards and an additional 20,000 in related industries. These facts alone are sufficient to justify and demand the perpetuation of our Merchant Fleet as carriers of passengers and freight. It is conservatively estimated that American companies in coast-wise, inter-coastal and foreign trades, expended directly a total of $100,000,000.00 for American labor and products during 1932.

The argument has been advanced that it is more economical to transport our products in foreign vessels than in those of our own registry. Those people who are in favor of such a plan will do well to remember that foreign flag-ships buy their supplies at home, not in America, and are manned by sailors of their own nationality. The only feasible method which we may employ to keep the American dollar at home is by fostering the upbuilding of an American Merchant Marine. This same idea was convincingly asserted by our progressive President, Thomas Jefferson, when he said, "For a navigating people to purchase its marine afloat would be a strange speculation, as the marine would always be dependent upon the merchants furnishing them. Placing as a reserve with a foreign nation, or in a foreign shipyard, the carpenters, blacksmiths, caulkers, sailmakers and the vessels of a nation, would be a singular commercial combination. We must, therefore, build them for ourselves. To force shipbuilding is to establish shipyards; is to form magazines; to multiply useful hands; to produce artists and workmen of every kind who may be found at once for peaceful speculations of commerce and for the terrible wants of war."

The Merchant Marine of any country owes allegiance first to the nation whose flag it flies. We have no right to expect the same loyalty from foreign ships as we would expect from American ships. Hence, it naturally follows that we cannot expect the same service from any foreign Merchant Marine as we could from our own. The Americans, meanwhile, continue to extend the bulk of our patronage to alien-flag carriers. We instigate more foreign commerce, yet we are carrying but one-third of what we ourselves help to create. We are the world's greatest travelers, yet three-fourths of our business goes to foreign lines. Shall our country in 1935 pay to British lines the gigantic sum of $94,000,000.00 in freight charges as was done in 1929, when British customers paid to American lines only $16,000,000.00 for similar services? In addition, American travelers spent $58,000,000.00 in passage money on British ships with no appreciable offset in the form of British payment to American lines. In this manner, we are contributing to the wealth of other nations rather than expediting the return of prosperity to our own nation.

The initial general fact is that the Merchant Marine is not solely or even primarily the business of the shipping industry or the foreign trade group; it is the business of the United States as a whole and its status affects the economic interest and vital welfare of every man, woman and child in the United States.

In conclusion, it may be truly said it is imperative that we have an adequate Merchant Marine whether by indirect or direct subsidizing, and in support of this assertion we quote so worthy an authority as Dr. Thomas H. Healy, assistant Dean, School of Foreign Commerce, Georgetown University: "It is a matter of economic and political life and death to us that, at all hazards, we must have a first-class Merchant Marine, protected and subsidized against increasing subsidized competition, which would otherwise drive it completely from the seas. The failure of our Merchant Marine would not only be a disastrous blow against our domestic, as well as foreign trade, but at the time when very real war threats fill the air, it would cripple our national defense and threaten our very national life."—"Marine Progress," July, 1935.

ASSOCIATION NOTES.

The following members have joined the Society since the publication of the last previous JOURNAL:

NAVAL.

Abbott, Royal, Lieutenant, U. S. Navy.

Blake, William A., Lieutenant, U. S. N., 270 Park Ave., New York, N. Y.

Bradley, Charles, 229 West 12th St., New York, N. Y.

Connell, George H., Lieutenant, U. S. N. R., 1001 4th St., San Francisco, Calif.

Craven, F. B., Mystic S. S. Co., Boston, Mass.

Harrison, Thomas J., Ensign, U. S. N. R., 1215 9th St., Port Arthur, Texas.

Snyder, Lee G., Ensign, U. S. N., care Oil Well Supply Co., Dallas, Texas.

William, Jack B., Lieutenant, U. S. Navy.

CIVIL.

Hobbs, Merrill H., Apartment 540, 1630 R St., N. W., Washington, D. C.

ASSOCIATE.

Gilchrist, James, Engineering Director, Barclay, Curle & Co., Glasgow, Scotland.